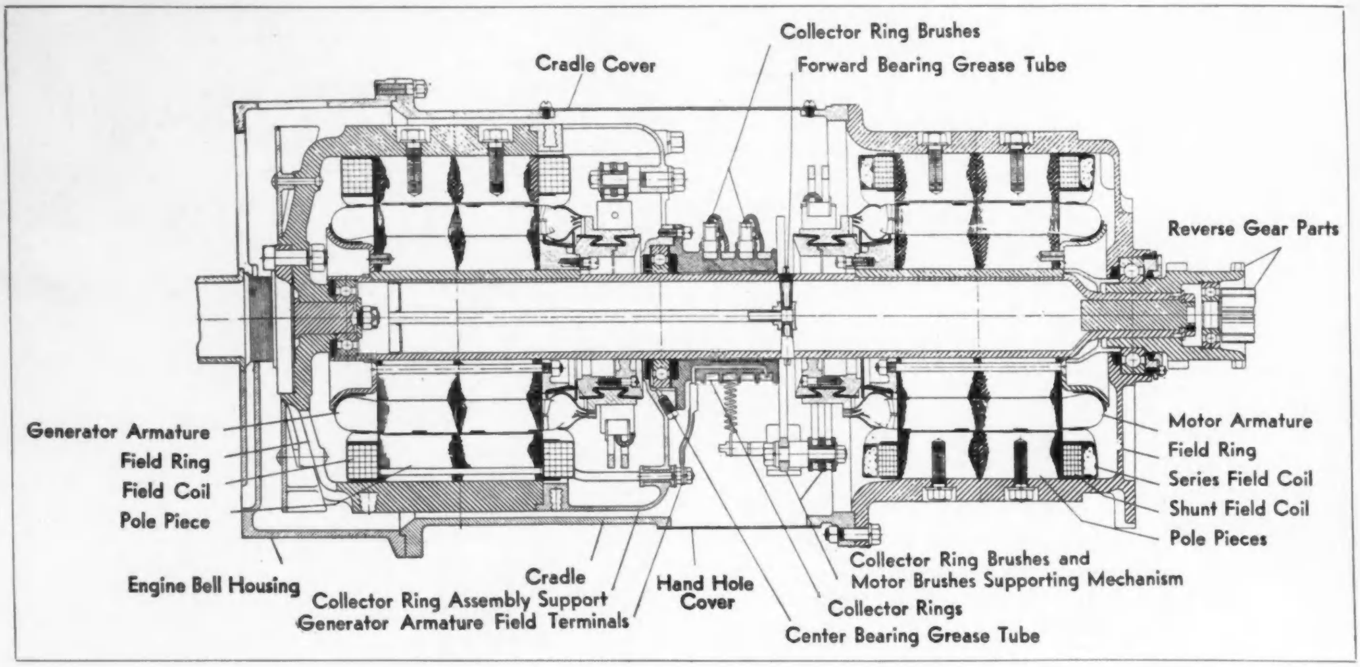


Fig. 2—Electric Transmission of the Owen Magnetic Car

recollection of a well-engineered, quality automobile that had much to recommend it. My friend, Edward H. Remde, who was chief engineer of the Baker R. & L. Co., Cleveland, which built the Owen Magnetic, comments recently on the car as follows:

"It had free-wheeling in a most ideal way but was brutal on the brakes. It had the ease of control that seems so desirable today, but it had the handicap of extra weight and cost. For example, the W-42 model weighed at least 600 lb. more and cost about $750 more than competing cars of conventional construction."

Judging from the foregoing, economic reasons seem to preclude its adaptation to the problem of automatic transmissions for the automobiles on today's market.

Another item which saw the light of day was the Cutler-Hammer electric gearshift shown in Fig. 4, used on the Premier car. Solenoids, as selected by the operator. were energized when the clutch pedal was depressed, which caused a plunger connected to the gearshifter fork to move to effect the shift.

The de Lavaud automatic transmission is illustrated in Fig. 5. Its advantages and disadvantages are stated elsewhere[1].

The Constantinesco torque converter[2] is illustrated in Figs. 6 and 7. The principle of operation is illustrated by reference to the action of a stick weighted with a ball at one end and suspended freely by a string at the other so that it can swing as a pendulum. Interference with the swing of the pendulum at any point on the stick results in a transference of part of the inertia force of the ball to other parts of the stick and also alters the amplitude and rate of vibration of all parts. At the same time, pressure is set up at the point of interference. The pressure varies in magnitude with the inertia and is proportional to the change of speed in a unit of time. In a motor-car, the engine, which develops a steady pull or torque at a constant speed, is used to drive against a resistance that requires either a variable torque or variable speed. The mechanical means employed to provide this variable torque can be reduced to the principle of the lever, in which varying the point of support changes the pressure required at one end to balance a constant weight at the other end. In practice in a motor-vehicle, this is accomplished by a change of leverage, or ratio, by shifting gears.

The Spontan automatic transmission or torque converter[3] is shown in Figs. 8, 9, and 10. In principle, it comprises two bob-weights connected to studs secured to the web of the engine flywheel by means of links. The bob-weights, 180 deg. apart, are connected rigidly to two eccentric straps surrounding two eccentrics on the main driving sleeve and are subject to centrifugal force, which varies as the square of the speed of rotation of the flywheel. This oscillating motion, through the double-roller clutch (Fig. 9), while moving in one direction, transfers its torque to the driven sleeve on the propeller-shaft and, when moving in the opposite direction, its torque is transferred to the outer or reaction sleeve. The driven sleeve carries a second flywheel to even-out the pulsating torque.

[1] See S.A.E. Journal, December, 1928, pp. 571-582; see also *Automotive Industries*, June 19, 1925.
[2] See The Journal, October, 1927, pp. 413-423.
[3] See *Automotive Industries*, April 12, 1930.

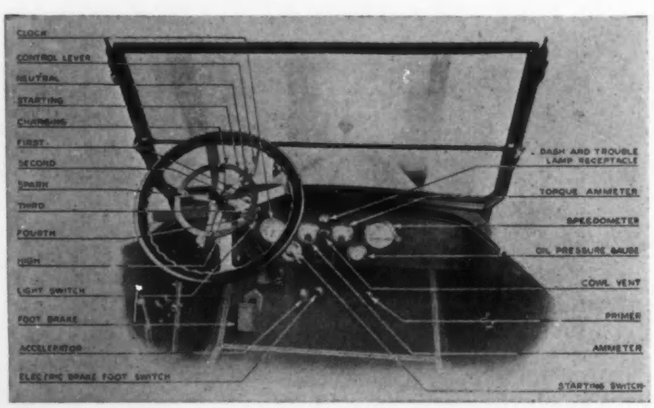
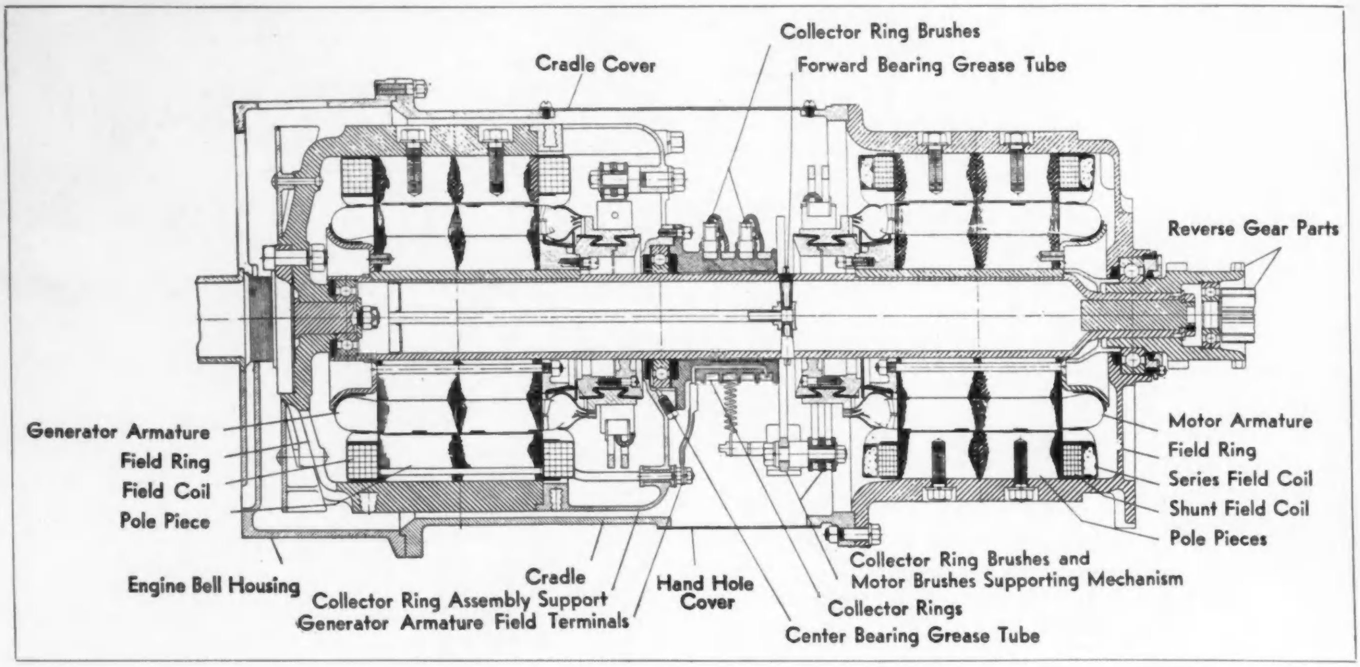


Fig. 3—Owen Magnetic Car Controls and Instruments



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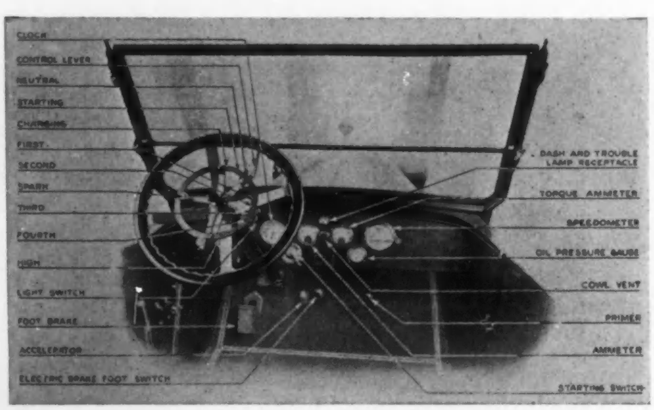


Fig. 3—Owen Magnetic Car Controls and Instruments

The Vickers-Coats hydraulic torque converter[4] is illustrated in Fig. 11. The primary element is attached to the crank-shaft and forms the rotating outer casing and a centrifugal pump, whose vanes are shown at *A*. The turbine, or driven-element, whose vanes are shown at *B*, is attached to a flange at the front end of the transmission-shaft. The third element, *C*, is stationary and has two sets of vanes, C^1 and C^2, the former being pivoted at their leading edges adjacent to the outlets from the turbine blades. They adjust themselves to the rate of discharge from the turbine, so that the liquid leaves the turbine blades without shock or impact and flows through the stationary element *C*, whence it is delivered through C^2 to *A*. The liquid used is "cutting compound." Direct drive is obtained by the dog clutch *D*, which is engaged by the spring shown. The torque is increased with increase of engine speed, since the delivery of liquid to the driven element increases. It is claimed that the output torque can be as high as four times the input torque, as in Fig. 12. The change of ratio that occurs with increase of torque is automatic, since increased resistance from the driving wheels slows the driven member *B* with respect to the driving member *A*. The transmission can be stalled by overload, but the engine cannot thus be stopped. At idling speed the engine will not drive the wheels. A separate reverse gear is required. If the dog clutch is engaged, the engine can be used as a brake; otherwise, it "free-wheels."

Allied to the hydraulic torque converter, and worthy of mention here, is the Salerni transmission or fluid flywheel, which is being presented to the automobile industry in this Country by De Xairer & Thomas and is now being used as standard equipment on the British Riley car. It is illustrated in Fig. 13 and consists of a driving member or impeller which, together with the housing, supplants the conventional flywheel. A driven member is splined to the main driving shaft of any conventional sliding-gear transmission, no synchronizing devices being used. A dog clutch is used at the rear end of the transmission, being arranged so that, after being disconnected by full movement of the clutch pedal, it is not reconnected until the driving end of it is brought up to a speed equal to that of the driven end. Ordinary motor oil is used, and rotation of the vanes *A* causes the liquid to impinge on vanes *B*, except when the cylindrical pressed-steel valve *C* is slid to the left so as to prevent circulation of the liquid. A slip curve of this transmission is shown in Fig. 14.

[4] See *Automotive Industries*, June 14, 1930.
[5] See *Engineering*, (London), June 21, 1912.

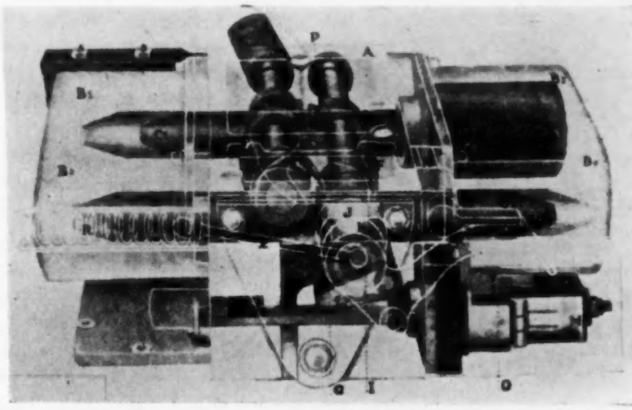


Fig. 4—Cutler-Hammer Electric Gearshift Used on the Premier Car

A—Gearshift Housing	*L*—Pawl Operating Master Switch
B—1-2-3-4—Coils	*M*—Master Switch
C—1-2—Magnet Cores	*N*—Locking Shaft
E-E—Camshafts	*O*—Master Switch Return-Spring
FF—Neutralizing Cams	*P*—Neutralizing Return-Spring
G—Ratchet Pawl Lever	*Z*—Neutralizing Return-Spring Shaft
I—Rocker Arm	
J—Operating Shafts	
K—Operating Lever	

Among its advantages are ability to crawl, stop or start in any gear without using the clutch pedal; improved acceleration; impossibility of stalling the engine; optional free-wheeling; the damping of torsional vibrations; less frequent shifting of gears; and the elimination of friction surfaces. The last point may be offset by the use of self-adjusting friction-clutches which have been developed. Other considerations in connection with hydraulic devices are the possibility of leakage, with serious consequences; heating of the liquid; variations in performance due to changes of temperature, with consequent changes in viscosity; problems in dynamic balancing; inefficiency; and difficulties in securing the correct ratios.

Before leaving the subject of hydraulic drives, the Hele-Shaw rotary pump and motor[5] should be noted as being typical of numerous brain-children of an earlier day. Fig. 15 shows the variable-stroke displacement-pump, which, as shown in Fig. 16, is connected by piping to the rear wheels, within which fixed-stroke motors, similar to the pump, are mounted.

The principles embodied in the Tyler automatic shift and clutch control are based on the provision, in an otherwise relatively conventional type of constant-mesh-gear transmis-

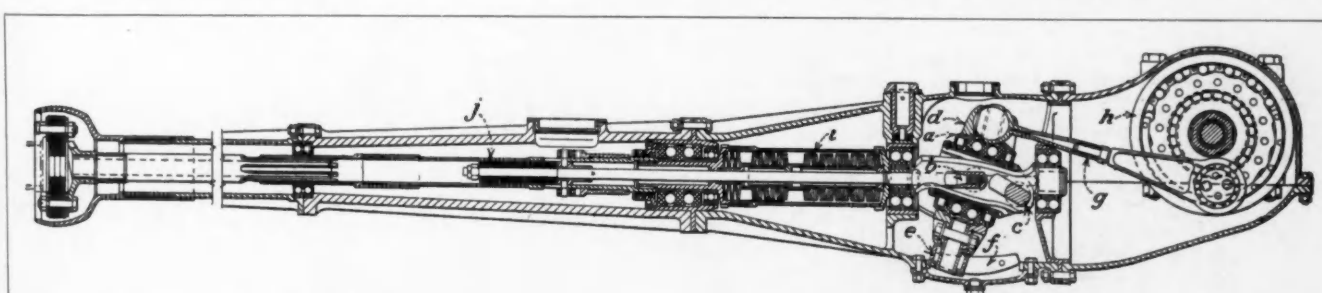


Fig. 5—Longitudinal Arrangement of De Lavaud Automatic Transmission, Partly in Section

The long transmission shaft is driven by the engine from the left end and rotates the inclined inertia-hub in the large section of the casing. Mounted on this hub is a non-rotating disc or ring supported on roller-bearings and carrying in circumferential sockets the spherical heads of six connecting rods. These rods are connected at their big ends to six roller-clutches or free wheels, which are mounted side by side on a sleeve that houses and drives the live rear axles of the car. Inclination of the hub and disc varies with variations in the engine torque and road resistance, which automatically varies the throw of the connecting rods and consequently the rate of rotation of the roller clutches, thus changing the car speed. A series of compensating spring-washers shown in the sectioned portion of the transmission shaft, cushions the reaction and gives a smooth continuous drive.

sion, of a means for clutching and engaging the torque-change elements; which requires no synchronization or time interval for actuation of the shift or selection. In other words, with the provision of a clutching mechanism that is both silent and instantaneous in its action, regardless of the number of revolutions per minute of the driving or driven members, it becomes a relatively simple matter to apply an automatic or power operating-means. A clutching means meeting these specifications having been developed, the obtainment of automatic shift required simply that a power shifting-device be supplied, the operation of which is controlled by the speed or torque, or both, of one of the driven members of a vehicle the engine speed of which is constantly proportional to the road speed.

Fig. 17 is a longitudinal vertical cross-section through the complete transmission, but not including the power-operating unit. For purposes of clarity, it may be considered that this view comprises four divisions of the structure; namely, at A1, the vehicle main-clutch throw-out mechanism; in the chamber A2, the gearing (referred to as the torque-change elements); at A3, the transmission-clutch unit (known separately as the Tyler) and at A4, the speed-sensitive control, or governor, for automatic shift.

The elements in division A1 comprise a synchronous detent for the vehicle main-clutch, together with a means for automatic lubrication of the clutch throw-out bearing with transmission lubricant for efficiency under automatic actuation. The clutch synchronous detent acts to restrain engagement of the main clutch—after it has been disengaged manually or automatically—until the engine has been accelerated to a speed equal to that of the clutch shaft (1). Of these elements, (2) represents a casing extending forwardly from the transmission to enclose the clutch throw-out bearing and detent elements for lubrication; the lubricant seal being provided by baffle grooves on the outside of a sleeve (3). The latter is rotably mounted on the shaft (1) but is fixed to the main clutch throw-out fingers (4) so as to rotate at engine speed at all times. Any rotation of this sleeve relative to the shaft (1) occurs only when the main clutch is disengaged. Therefore a bearing (5) for the sleeve on the shaft (1) provides in itself a suitable lubricant seal at this point.

The rear face of sleeve (3) is provided with an enlarged diameter in which are formed two spiral or angle-faced extensions (6) upon the outside of which is mounted the clutch throw-out bearing (7). Adjacent to the spiral extensions are a pair of rollers (8) affixed by studs to a collar (9). The

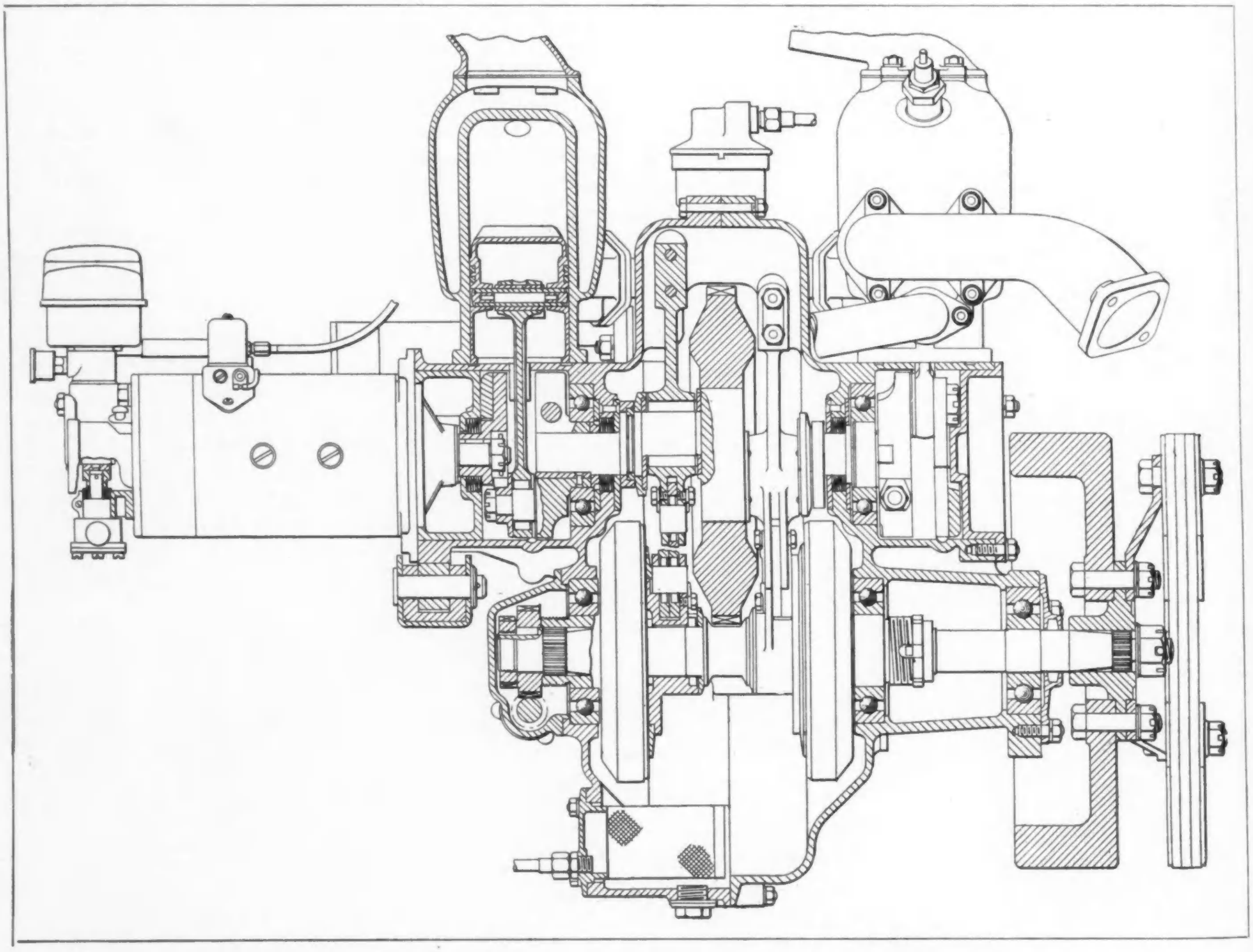


Fig. 6—Constantinesco Powerplant and Transmission in Longitudinal Section

The torque-converter flywheel is carried in the center on the engine crankshaft and two short cranks or eccentrics on the shaft oscillate the two inertia elements on either side of the flywheel. Below is the secondary shaft, which is driven by the "mechanic valves" and in turn drives the propeller shaft through a flexible coupling at the right.

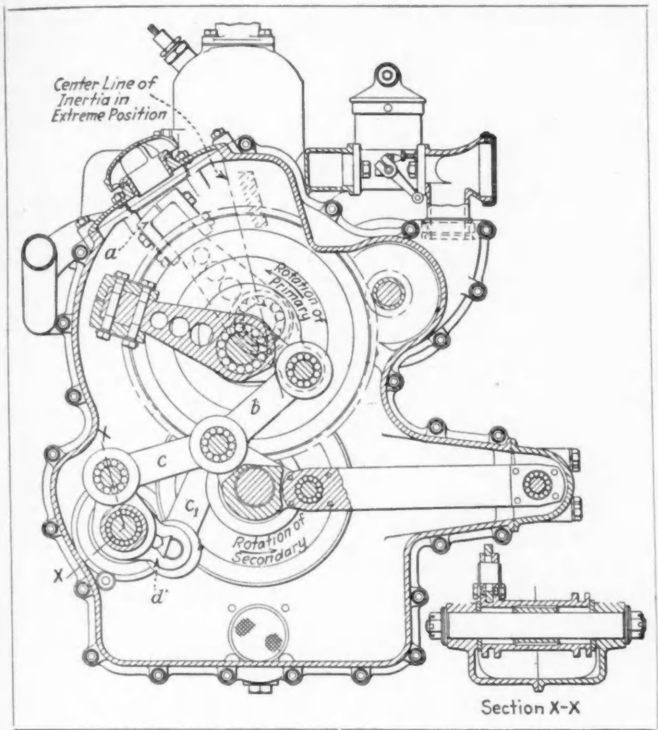


Fig. 7—Torque-Converter in Transverse Section

This is the construction as embodied in the Constantinesco 5-hp. car. The transmission is disposed between the two engine cylinders, and the engine crankshaft is the primary shaft of the torque converter. Its rotary motion is transmitted to inertia weight *a* and is converted into reciprocating motion by the short arm of the inertia lever. Connecting link *b* then transmits the impulses through rods *c* and c_1 to the "mechanic valve," or roller clutch d_1, reconverting them into rotary motion to drive secondary shaft.

latter is also rotatably mounted upon the shaft (1), but is adapted for only a limited rotation relative to sleeve (3) between the highest and lowest points of the spiral extensions. When the vehicle main-clutch is disengaged and the engine speed reduced to less than the speed of shaft (1), the rollers (8) lie opposed to the high points of the spiral extensions (6). With the rollers in this position when the clutch bearing is released for engagement, the highest points of the spiral abut against the rollers (8) in the collar (9), which latter in turn thrusts against the face of a second collar (10), the latter being fixed to the shaft (1) by lock balls (11). When the engine speed has been advanced to equal or slightly exceed the number of revolutions per minute of shaft (1), the opposing rollers (8) are urged down the inclined face of the spiral extensions (6) and in this manner permit engagement of the clutch at the moment of relatively substantial synchronization of the speeds of the driving and driven members.

The synchronous detent mechanism, of course, need not be in effect when the clutch is operated manually. Therefore, the same elements which are used to cut out automatic clutch-actuation are connected with a means for moving a lock-collar (12) rearwardly to release the lock balls (11), and hence the collar (10). This permits the collar to slide back along the shaft to an inoperative position relative to the highest points of the spiral extensions of sleeve (3).

In division A2 is shown the gearing. The gears are all of the helical type and are all ball and roller-bearing mounted. The type of mounting employed presents great rigidity due to the fact that the supporting bearings (13) and (14) are exceptionally close together. The jackshaft group (15) constitutes a compact cluster on a short shaft, practically precluding the possibility of deflection. Gears (16) and (17) extend through the bearing (14) in sleeve form, terminating in short splined ends to which are fixed the clutches.

Reverse is provided in the form of a conventional sliding gear, a portion of which is shown at (18), this being adapted for engagement with gear teeth (19) surrounding the transmission-clutch unit. Gear (18) is mounted upon a short splined shaft, not shown, which, at the front end, is provided with a helical gear in constant mesh with the small gear (20) of the jackshaft group. At the top of the gear chamber is shown the end (21) of a conventional shift lever. The structure is designed so that this lever will act either for direct manual shifting, if desired, or as an over-control for the automatic shift. A ball link (22) affixed to the throw-out fork (23) of the transmission clutch-unit acts in conjunction with lock plungers (24) operated by high points on the shifter rails to lock out both the vehicle main-clutch and transmission clutch when the shift is made to neutral. This is a feature that is advantageous where the operator forgets to hold the clutch disengaged while starting the engine in cold weather, as is good common practice.

Operation of Tyler Transmission Clutch

Divisions A3 and A4 of Fig. 17 will be understood better by reference to Figs. 18A and 18B which illustrate a general

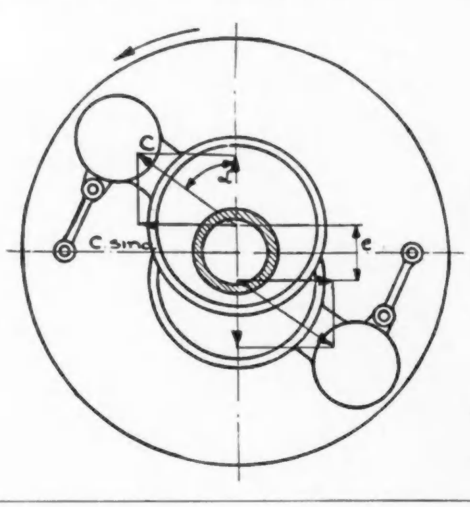


Fig. 8—Diagram Showing Inertia Weights and Eccentrics of the Spontan Automatic Transmission

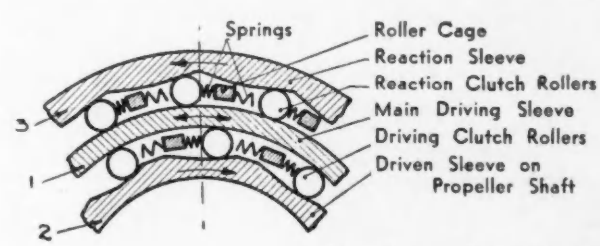


Fig. 9—Diagram of the Spontan Double-Roller Ratchet Clutch

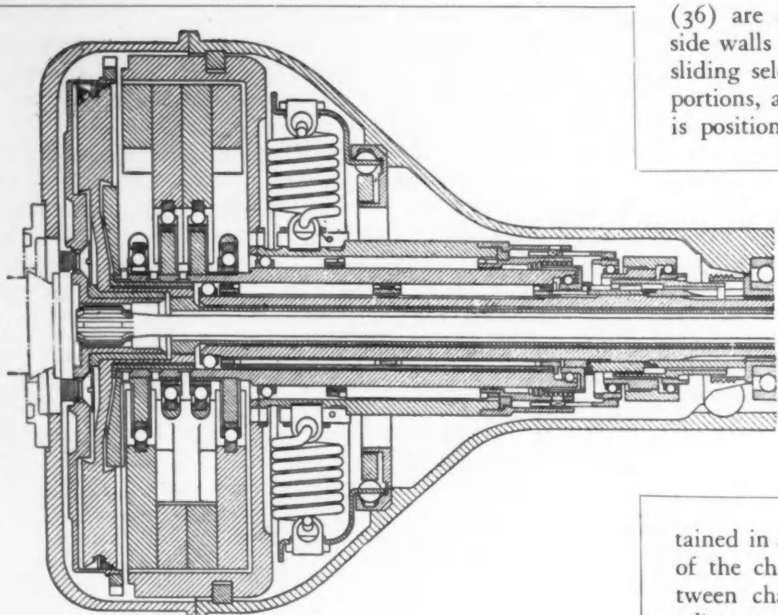


Fig. 10—Longitudinal Section through the Spontan Transmission

plan of the complete operating system in semi-diagrammatic form. Fig. 18A represents the transmission clutching unit. Numerals (25), (26) and (27), indicate the sleeves and shaft extending from the gears referred to in Fig. 17. The splined ends of these members are engaged with internally toothed discs (28) of the selectable clutch-unit. One sleeve engages directly with the toothed discs and the others are provided with auxiliary collars (29) and (30). The clutching unit contains three independent clutching sections, (31), (32) and (33), each having a torque capacity more than sufficient for the torque ratio of the particular grouping of driving gears selected.

The pressing of the clutch for engagement is provided by springs (34), acting against a main pressure-plate (35), which, acting in connection with channels (36), transfers its pressure to one of three sub-pressure plates, (37), (38) and (39), previously selected in a manner to be described. Three such channels are provided. Slidably disposed in channels (36) are selector fingers (40), and loosely retained in the side walls of the channels are three pairs of balls, (41). The sliding selector-fingers are formed with narrow and widened portions, as shown. When any one of the widened portions is positioned between any pair of balls, the latter is maintained in a position projecting outwardly from the side walls of the channels, the projecting portions forming a lock between channels (36) and the particular sub-pressure plate adjacent to that point. Upon release of the clutch pressure-springs, the channels act as drawbars between the main pressure-plate (35) and the particular sub-pressure plate selectively engaged therewith. A neutral position is also provided where none of the widened portions of fingers (40) are between any of the pairs of balls. In this position no sub-pressure plates are acted upon with the release of the clutch pressure-springs and a non-driving relation is thereby established.

The actual release or engagement of the transmission clutch-unit, after selection, is effected with the same elements that control the main clutch of the vehicle. The operating connections are arranged, however, so that the transmission clutch-unit is fully engaged or released before engagement or release, respectively, of the main clutch. In other words, the transmission clutch-unit is never engaged under load and hence does not transmit torque until after *full* engagement. In this manner the transmission clutch-unit picks up only the load of the free-running elements of the transmission. The designer states that, under these conditions, examination and check after 30,000 miles of operation has shown the clutch-

Fig. 11—The Vickers-Coats Hydraulic Torque Converter

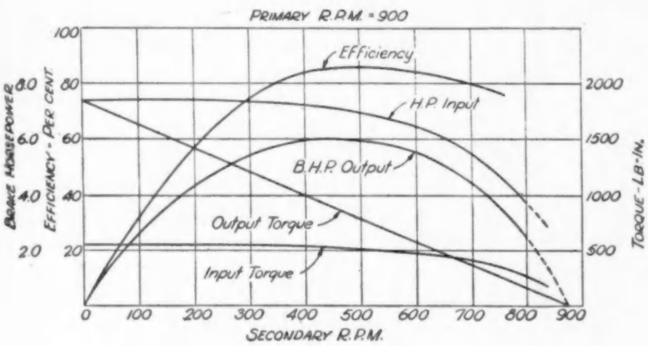


Fig. 12—Characteristic Curves of the Vickers-Coats Torque Converter

plate wear to be so infinitesimal as to be practically non-measurable.

Fig. 18B is a vertical longitudinal cross-section through a combination clutch power-actuator and automatic or power-shifting unit. The automatic system in general comprises a governor (42), illustrated diagrammatically, which, relative to the speed of the vehicle, transfers its movement to a valve actuator (43), the latter being connected by a pin (44) to a sliding valve (45). A series of ports (46) are adapted to be opened individually to suction by varying the positions of the sliding valve (45), influenced by the governor. The suction connection through the various ports acts upon a series of diaphragms (47) to cause a plunger (48) to assume any one of four definite stopping positions, these being transferred to a conventional shifter rail (49), causing it to define the shift positions of "first," "second," "high" or "neutral." The connections from the governor to the valve actuator and from the plunger (48) to the shifter rail are illustrated diagrammatically by dot-and-dash lines.

Movement of the valve actuator (43) in response to the governor is controlled by means of detents (50) which work in conjunction with a double-acting compression spring (51) so as to cause the sliding valve (45) to move instantly from port to port, instead of gradually, as relative to the governor action. This prevents the valve from lingering at any time between full port-openings. The valve (45) is made of hard chromium-plated brass and operates without lubrication. After the valve (45) has been positioned by the speed of the vehicle as described, the actual shift does not occur until a main valve (52) opens the system to suction through a small manifold—not shown—adjoining the ports at one side of the sliding valve (45). The main valve (52), as shown diagrammatically at (61), is connected to one of the clutch-operating elements, indicating that the shift occurs simultaneously with disengagement of the main clutch and the transmission clutch.

A set of spring plungers (53), operated upon movement of the main valve (52), provide an escapement for the valve actuator (43) to prevent the latter from moving past the position of "second gear". This escapement allows the valve to move to "high-gear" position upon the next release of the clutch after the "second-gear" position has operated. A partition (54) separates the power shifting-unit from the clutch power-actuator. The latter is operated independently through actuation of a control valve (55). The entire system is controlled by the operator with a master control-button situated adjacent to or within, but separate from the accelerator pedal. This construction provides the operator with means for restraining both free-wheeling and shifting at all times, since the shift as well as the clutch power-actuation is dependent upon full retraction of the master control-button. The latter projects somewhat above the accelerator pedal, thereby permitting the throttle to be fully closed without complete retraction of the master control-button, thus restraining its action. The arrangement of the accelerator pedal and the master control-button is illustrated diagrammatically in plan and in side view at (56).

An independent port (57) opened or closed by the sliding valve (45) is provided to form an auxiliary bleeder-path for the clutch power-actuator. This opens only when the vehicle has reached a predetermined speed and thereafter increases the normal bleeding action of the clutch power-actuator sufficiently so that clutch engagement becomes substantially controlled by the main-clutch synchronous detent. A shift lever (58), shown diagrammatically, illustrates the manner in which such a lever is used to give the operator personal control at any time over the shift or selection automatically caused by the governor. Only one suction connection is required between the power-operating unit and the intake manifold, as shown at (59); also, but one opening is required for admission of atmosphere to the system, as shown at (60).

Advantages Claimed for the Tyler Transmission

It is claimed that the system in general provides the following features:

(1) The shifting and selector elements are such that no sound can occur during shifting at any speed of the vehicle.

(2) The shift is instantaneous in its action upon release of the main and transmission clutches.

(3) The automatic or power shift remains effective upon manual disengagement of the clutches if the clutch power-actuator, or "free-wheeling", is shut off.

(4) The combination of the power shift-unit with the clutch power-actuator, together with all operating valves, eliminates excessive piping and other extraneous connections.

(5) The main-clutch bearing is automatically lubricated

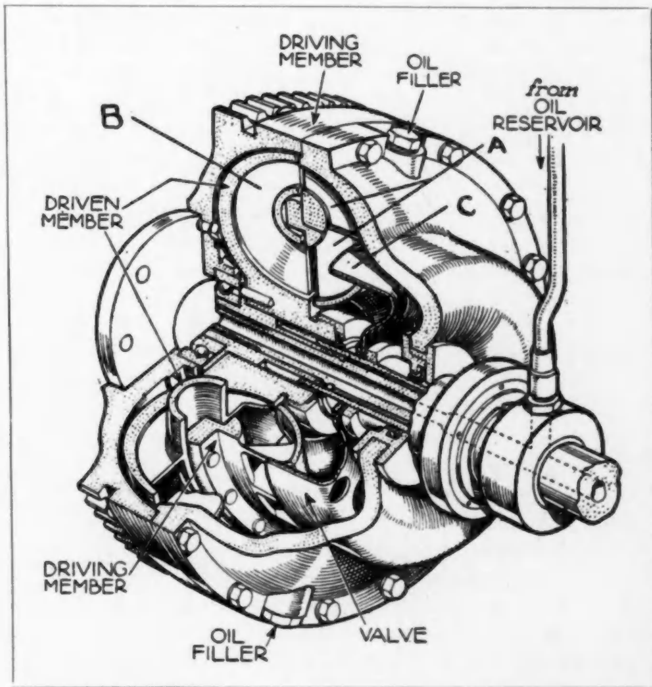


Fig. 13—Various Features of the Salerni Fluid Flywheel

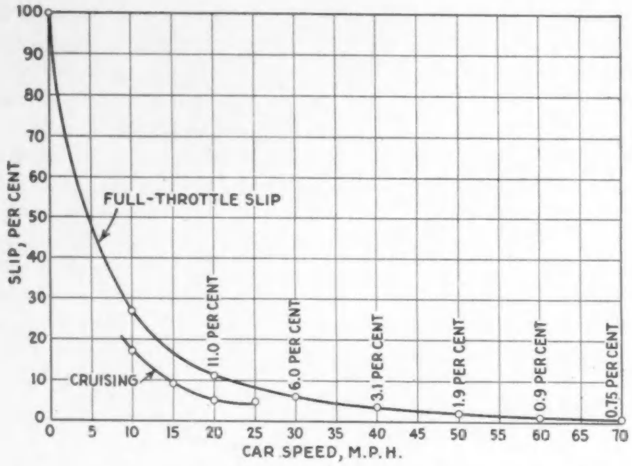


Fig. 14—Typical Slip-Curve for the Salerni Hydraulic Torque Transmitter

by the transmission lubricant for efficient automatic clutch-actuation.

(6) The main-clutch synchronous detent provides for rapid engagement of the clutch under power actuation without lurch, and eliminates excessive wear of the clutch disc.

(7) The speed-controlled bleeder-valve opening, working in conjunction with the clutch synchronous detent, eliminates any possible lag in the vehicle acceleration.

(8) The gear construction provides the utmost in rigidity for the mounting of helical gears.

(9) Free-wheeling, as well as the occurrence of shifting, is at all times under instant control of the operator by means of the master control-button.

(10) Manual shift or over-control of the automatic shift is at all times instantly available to the operator, by means of a small shift-lever projecting through an H-plate mounted just below the steering-wheel.

The Mono-Drive is a planetary spur-gear system developed by Oscar H. Banker of Chicago and effects the three forward speeds through several automatic clutches. The operator moves an "engager" which extends from the instrument board to select *forward, neutral* or *reverse*. When in the *forward* position, an automatic main clutch engages at about a 500-r.p.m. engine-speed and starts the car in low gear, which is effected through the planetary gear-system. Above a speed of about 12 m.p.h., the second-speed automatic-clutch engages when the operator raises his foot from the accelerator pedal for an instant. Drive will continue in second gear up to any speed so long as the engine is driving the car. When the throttle is closed sufficiently to allow the engine speed to fall below that of the propeller shaft, thus reversing the torque through the transmission, direct drive is established through a third automatic clutch which is of the expanding-spring type.

Upon decelerating, the high-speed clutch disengages at a speed of about 10 m.p.h. and effects instantaneous shift to second gear, regardless of throttle manipulation. The second-speed clutch disengages at a speed of about 15 m.p.h. in second speed and effects instantaneous shift to low gear without regard to throttle manipulation. The main clutch disengages at a speed below 5 m.p.h. in low gear, provided the throttle is closed.

When the car is stopped, the transmission is in low gear, and declutched.

When the engager is moved to neutral position, the reaction member of the planetary system is released from the case.

When the engager is in reverse position, the second-speed drive-pinion is held stationary by being locked to the main housing, thus causing the planet gears to revolve around the main axis in a reverse direction.

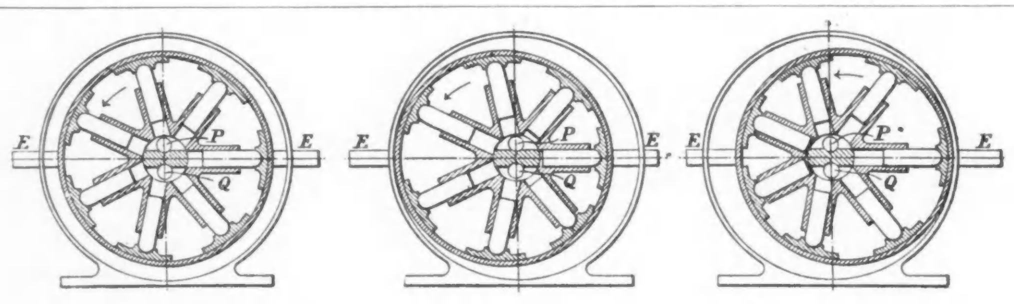
Fig. 19 is a longitudinal section through the Mono-Drive. Fig. 20 illustrates the torque flow under the six conditions of (1) low gear, (2) second speed, (3) high gear, (4) changing from high gear to second speed, (5) reverse drive and (6) neutral position.

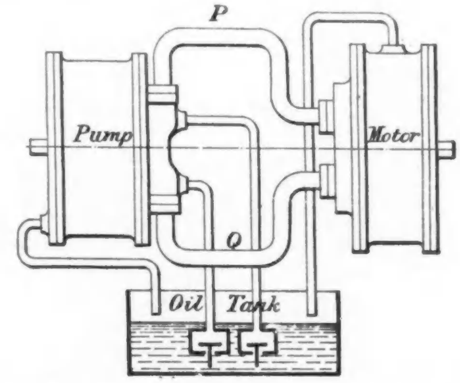
Fig. 21 shows the installation in a car and that the general dimensions are not appreciably different from those of a conventional transmission. The small pedal at the left of the brake pedal can be depressed at any time to effect an instantaneous shift from high to second speed.

Free-wheeling is possible in low or in second gear to the extent that the engine may slow down to car speed, as if in high gear. A pull button on the instrument board actuating a Bowden wire when pulled out, makes effective two positions of the second-speed pedal; pushed its full stroke and held there, second speed is maintained regardless of throttle manipulation; at half stroke, semi free wheeling occurs in coasting.

Fig. 22 shows a centrifugally acting automatic clutch, and Fig. 23 shows the planetary assembly and automatic expand-

Fig. 15—The Hele-Shaw Rotary Pump and Motor, Typical of an Hydraulic Drive of an Earlier Day





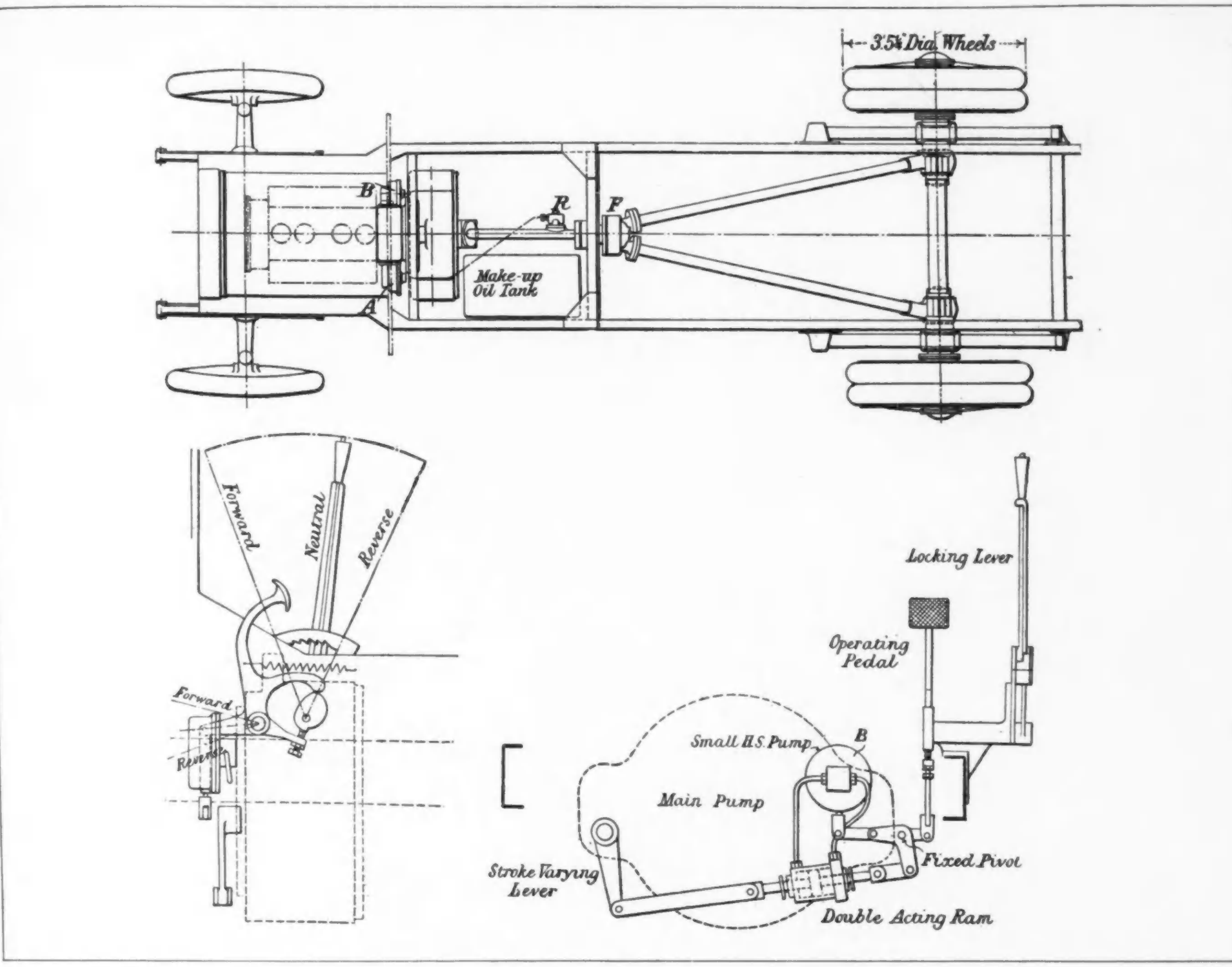


Fig. 16—Variable-Stroke Displacement-Pump of the Hele-Shaw Hydraulic Drive

ing spring-type clutch which is attached to the planetary carrier.

The Mono-Drive has the advantage of being all-mechanical, no vacuum, pressure or electricity being necessary to its operation. The automatic clutch is not subject to gradual release, as it disengages instantly. It is provided with an adjustment as to speeds of cutting in and out, to suit the preference of individual drivers. It has a No-Rol-Bak feature. All speeds can be obtained by manual operation. The inventor states that a car equipped with the Mono-Drive has been operated during the last two years for more than 45,000 miles with practically no changes in the mechanism.

It is commonly stated that the best fuel economy can be obtained by operating with wide-open throttle. An exploration of the possibilities in this connection, assuming a transmission of an infinite number of ratios, is thought to be in order.

Fig. 24 shows the fuel consumption of a six-cylinder engine of 3⅜-in. bore and 5-in. stroke under full, ¾, ½ and ¼ throttle-openings. The data were taken from test results supplied by Prof. W. E. Lay, of the University of Michigan. The curves clearly show that, for each of the four speeds shown, the fuel consumption per horsepower hour increases as the throttle opening is reduced.

Fig. 25 presents data, furnished by L. P. Kalb, referring to a Continental engine, which I have shown applied to a car having a 3.9:1 rear-axle ratio, and also to a speculative car. The horsepower consumed by chassis friction is represented by Curve *OA*, the horsepower consumed by chassis friction plus wind resistance is represented by Curve *OB*. The horsepower available at the flywheel under full throttle is represented by Curve *OC*. The intersection of Curves *OB* and *OC* shows the maximum speed of the vehicle in direct drive to be 82 m.p.h. Fuel consumptions and horsepower are shown also for full, ¾, ½ and ¼ throttle-openings.

Let us now consider a speculative car operating with full throttle-opening, on the assumption that the driver can as readily avail himself of an infinite number of gear ratios, within certain limits, as he can produce an infinite number of throttle positions in today's conventional car. Curve *OC* now relates only to engine speed, while Curve *OB* relates to car speed. From Curve *OB*, 27 hp. is required to maintain a speed of 55 m.p.h., level pavement and still air being assumed. The horizontal line representing 27 hp. intersects Curve *OC* at *F*, which means an engine speed of about 950 r.p.m. and a fuel consumption of 0.61 lb. per hp-hr., requiring an overall gear-ratio of approximately 1.5:1. In the conventional car the engine speed would be 2480 r.p.m. with throttle open and a fuel consumption of about 0.79 lb. per hp-hr., an increase of about 30 per cent. It will be seen that the 27-hp. line also crosses the speed-horsepower curves of ¾ throttle; also that the fuel consumption for this opening

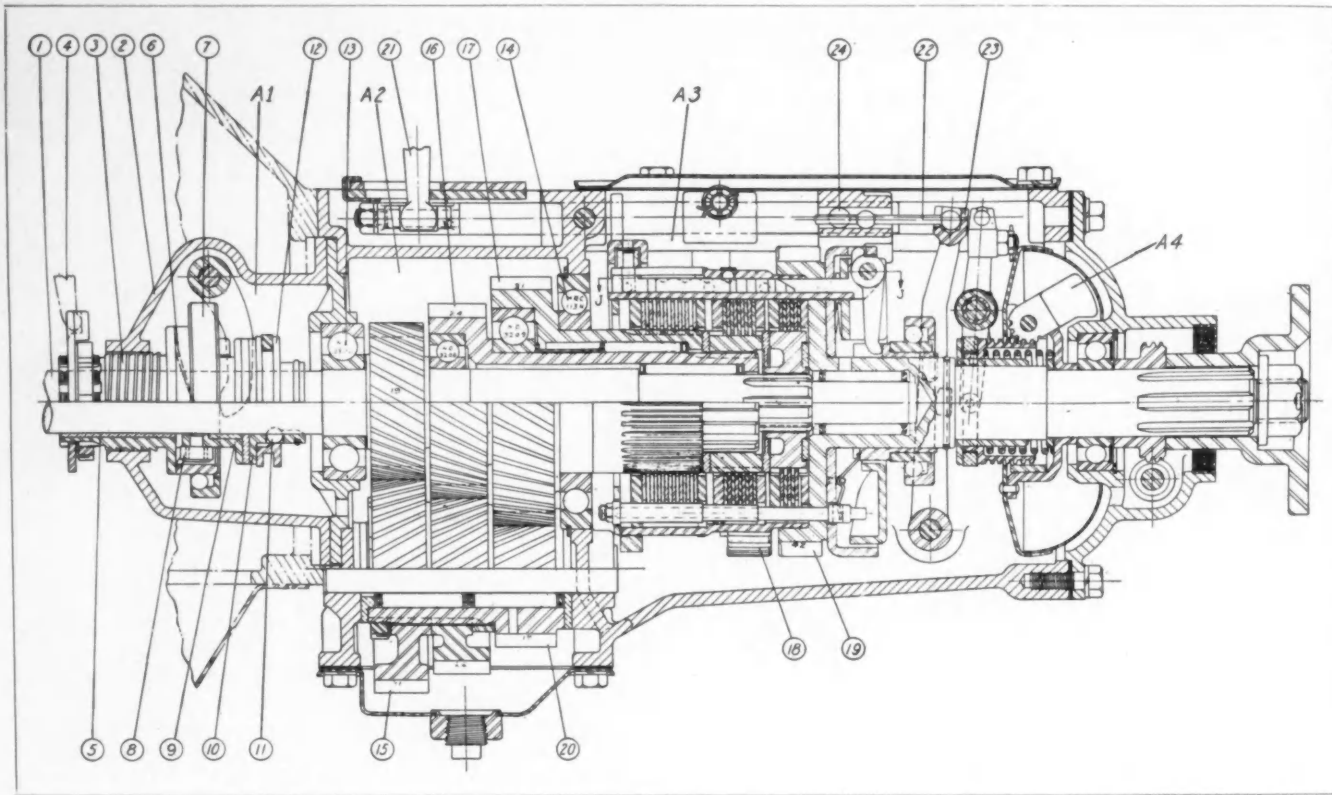


Fig. 17—Longitudinal, Vertical Section Through the Complete Tyler Automatic Shift and Clutch Control, But Not Including the Power-Operating Unit

would be 0.59 lb. per hp-hr., which is slightly better than with full throttle.

At 30 m.p.h. the required power is only 8 hp. which, at full throttle, occurs at 343 r.p.m., with a fuel consumption of 0.84 lb. per hp-hr. The conventional car would require an engine speed of 1358 r.p.m. at less than ¼-open throttle, with fuel consumption of at least 1.05 lb. per hp-hr., an increase of 25 per cent. The 8-hp. line crosses the remaining three speed-horsepower curves and shows 0.86, 0.85 and 1.04 lb. per hp-hr. for the ¾, ½ and ¼ throttle-openings respectively.

The vertical distance between Curves *OB* and *OC* represents at all car speeds the horsepower available for acceleration in the conventional car with 3.9:1 overall ratio. In the speculative car, operating at full throttle, the horsepower available for acceleration would be the vertical distance between Curve *OB* and the 75½-hp. line (maximum for this engine); since, with the throttle already wide open, the operator has only to effect an overall gear-ratio that will allow the engine to turn at its maximum-horsepower speed. A governor would then be required to alter the ratio throughout the acceleration, keeping the engine speed constant. Such an arrangement will produce greater acceleration than is now possible with any conventional car.

Fig. 26 illustrates the relation in the speculative car between car speed and (*a*) engine revolutions per minute, (*b*) overall gear-ratio to maintain any speed, and (*c*) overall gear-ratio to secure maximum acceleration.

A conventional car in which the maximum car-speed occurs at an engine speed beyond that of maximum horsepower could, by means of an infinitely variable ratio, have its maximum speed increased by maintaining maximum horsepower, during acceleration, up to the point where Curve *OB* in Fig. 25 intersects the line of maximum engine horsepower. In the case of Fig. 24, this would effect an increase of only ½ m.p.h. in the top speed of the car.

In addition to possible economy in fuel consumption resulting from lower engine-speeds, the saving in lubricating oil, as is evident in Fig. 27 furnished by Ralph Teetor, should not be overlooked. When driving at high speeds, particularly in older cars, it is frequently necessary to stop more often for oil than for gasoline, a point which the owner will not forget. It seems unfortunate that both the studies along the line of gear ratios and along the line of ideal streamlining do not seem to offer possibilities of good fuel-economy at low car speeds.

Present-day four-cycle engines cannot be expected to operate satisfactorily with full throttle. This suggests that further study along the line of gear-ratio change effected by engine torque, as taught in recent patent art, may yield worthwhile results. A transmission with one range of ratios up to 35 m.p.h., and an automatically varying range controlled by pressure on a two-part pedal, such as used in the Tyler unimatic shift and clutch control, may point the way toward a more ideal solution of the problem by the use of three or four fixed transmission-ratios.

Relation between Time and Speed

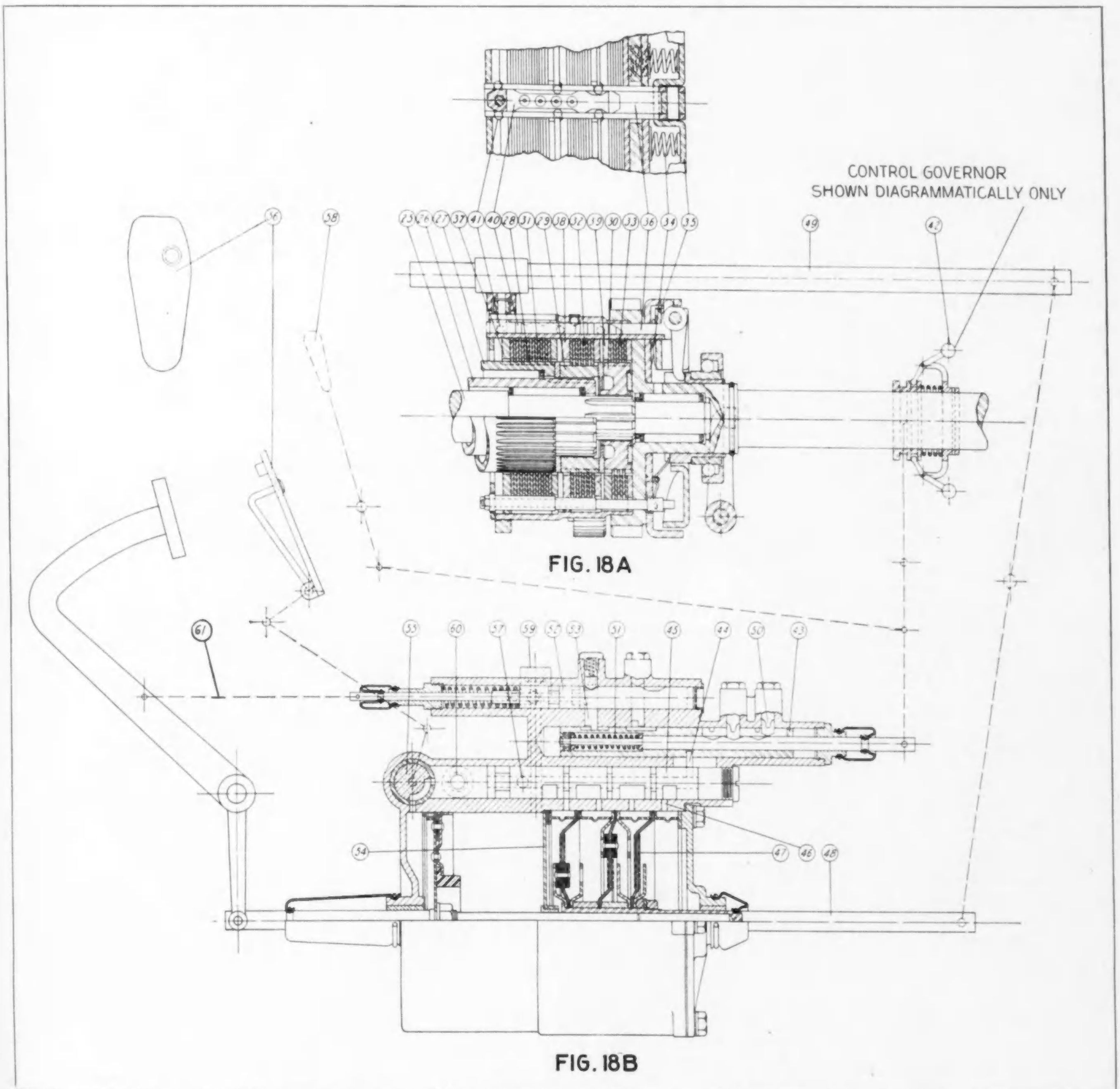
Fig. 28 shows, for a late-model car, the relation between time in seconds and car speed during acceleration through low, second and high gears. The car actually decelerates during the shift interval. If no interval were necessary, a precious second or so could be saved, as suggested by the dashed line, in addition to the possible improvement resulting from operating the engine at peak horsepower, which would effect a still greater saving.

The design of transmissions of the future doubtless will

have to take into account the need for real fuel economy and satisfactory car performance at sustained speeds much in excess of the peak speeds of most present-day automobiles, as suggested in Fig. 29. This is the composite of the opinions of such experienced men as C. W. Spicer, Lyle Snell, H. H. Rice, George McCain, Walter Fishleigh, Walker Gilmer, Prof. W. E. Lay, and others. It shows the average cruising speed of the public with modern cars, on the best highways under the best driving conditions. I believe all will agree that it is a conservative representation of the past and present. It forecasts cruising speeds of 70 m.p.h. in 1942 and 100 m.p.h. in 1952, provided that the highways are improved to permit, the tires are improved to provide adequate stamina, and drivers become sufficiently educated. Automobiles are now capable of sustaining speeds far beyond those which our highways safely permit. If highway improvement does not progress, our cruising speed (average) may be limited to about 55 m.p.h., as suggested by the lower dashed line, and progress in inter-city speed may then be no more rapid than has been the case with the railroads in the last 30 years.

Reactions of Operators to Automatic Gearshifting

In the summer of 1932 I witnessed the operation by a 12-year-old girl, who had never before driven any vehicle, of an automobile equipped with an automatic gearshift. Although her judgment was immature, she was able to start, accelerate through gears, and stop the car with less than five minutes' instruction and with considerable pleasure. An older woman, who is an experienced driver but has no mechanical knowledge, evinced the greatest pleasure when shifts



Figs. 18A and 18B—Semi-Diagrammatic Arrangement of Tyler System

into second and high gear were automatically completed without her having to operate with that mysterious and hateful contraption known as the gearshift lever.

Another middle-aged woman was at first thrilled with the novelty of the automatic gearshift; but, after considerable driving she preferred an easy, convenient and sure manual gearshift, enjoying her skill in controlling a powerful motor car just as she preferred controlling a spirited horse by holding the reins in her hands, as contrasted with riding behind an old nag with the reins wound around the buggy whip.

An engineer friend of mine, who says that the most important control of his radio is the shut-off, also says that the most important control of an automatic shift would be the shut-off, since in emergencies it must occasionally be necessary to stay in, or out, of gears contrary to the shift which would automatically be made.

A large number of engineers express the opinion that, since an automatic gearshift cannot think and the driver must therefore resort at least occasionally to manual control, it seems uncalled for to encumber an automobile with the added complication and probable added weight and cost, when our present manual gearshifts are so highly satisfactory.

Future of Automatic Transmission

A problem clearly stated is generally more than half solved. No group of men in the world could, 30 years ago, have produced a car as good as those built in America today in the lowest price-range, largely because the problems that have since been solved were not then clearly stated. Today, as yesterday, motorists have unfilled or partly-filled wants which I believe are: (*a*) economy, as witnessed by the relative sales of low-priced and high-priced cars; (*b*) safety; (*c*) com-

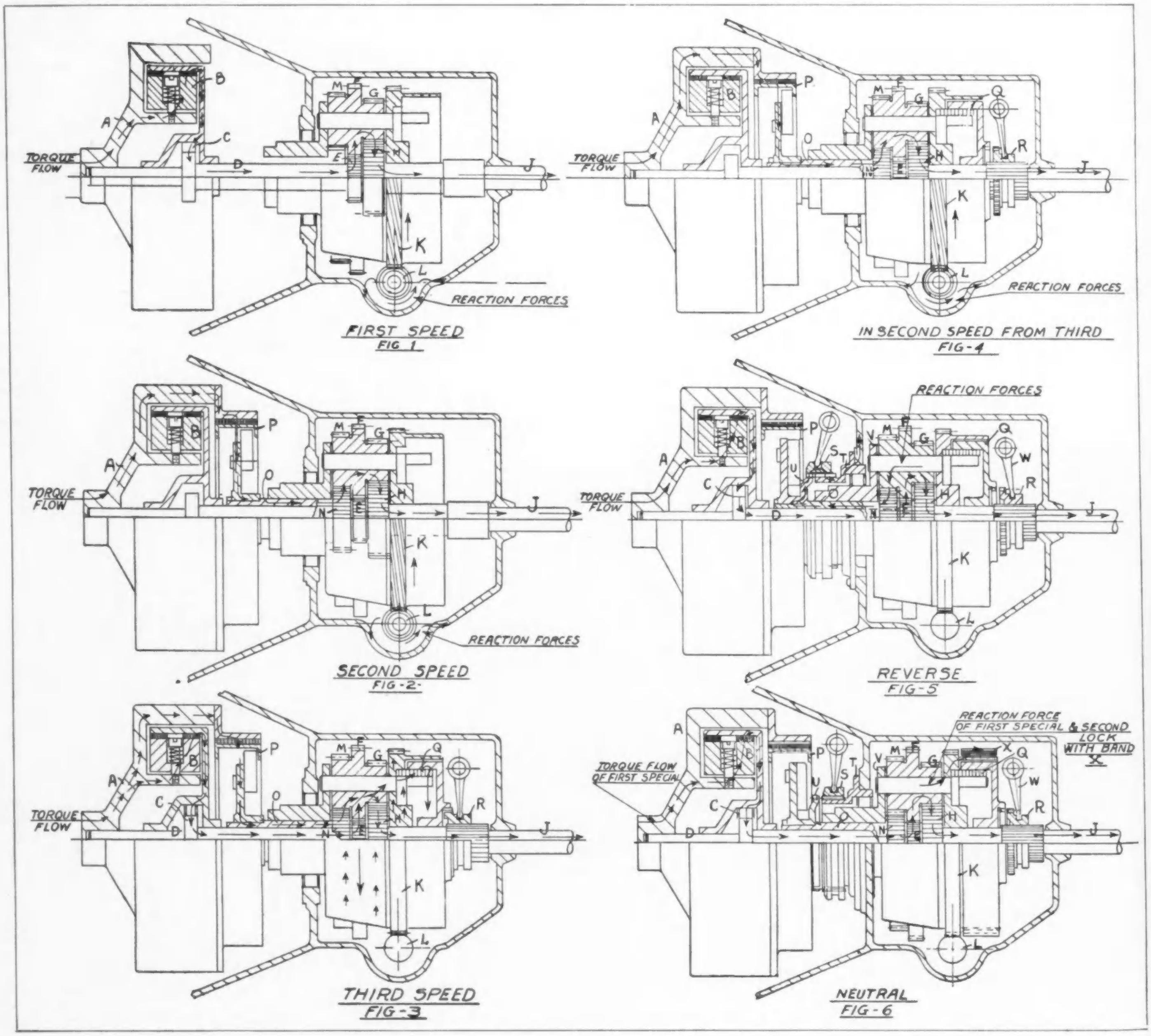


Fig. 20—Torque Flow of the Mono-Drive Transmission under the Six Conditions of (1) Low Gear, (2) Second Speed, (3) High Gear, (4) Changing from High Gear to Second Speed, (5) Reverse Drive and (6) Neutral Position

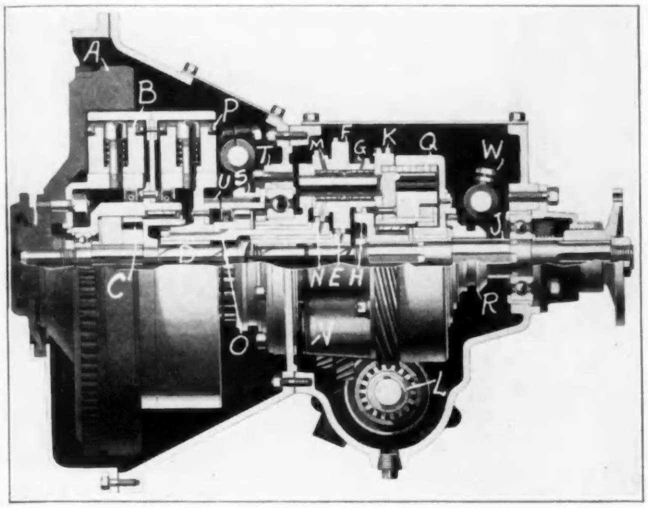


Fig. 19—Longitudinal Section Through the Mono-Drive Transmission

fort; and (*d*) speed, satisfactory appearance being assumed. Automatic transmissions can contribute to each of these except safety, and they constitute one of our present problems, which I have endeavored to state; partly, at least.

Every manufacturer is looking for sales advantage over his competitors. At present, we have opinions only as to whether the automatic transmission will constitute a sales advantage. Very recently we saw four-speed transmissions offered to the American public. Some companies have since gone back to three-speed transmissions. While I believe that any good car giving exceptional gasoline mileage will outsell its competitors who do not give half as much mileage, the fact that, in many instances at least, drivers cared nothing for the, to them, unmeasurable fuel economy which could be obtained through a four-speed transmission and were too shiftless to shift, seems to point very strongly in the direction of automatic transmissions, particularly since by far the greater amount of driving by the American public each year is under conditions where automatic shifts, or automatically varying gear-ratios, if of adequate stamina, could be made to function safely and satisfactorily without interference from the driver.

There is much activity at present in connection with the development of automatic transmissions. If the public wants them, they will appear as standard equipment. I believe that a good automatic transmission will appeal to the American public. Our company is now preparing to offer an automatic transmission at retail for 10 modern cars, and public interest in this connection will be watched with keen attention by car manufacturers and parts companies.

I consider that the desirable features of an automatic transmission are (*a*) reliability; (*b*) quiet operation; (*c*) reasonable cost; (*d*) reasonable weight; (*e*) reasonable simplicity; (*f*) reasonable efficiency; and (*g*) correct functioning which means that it must do the right thing at the right time.

I believe that automatic transmissions will come into use before streamlining, and that they will constitute the next important sales feature on automobiles.

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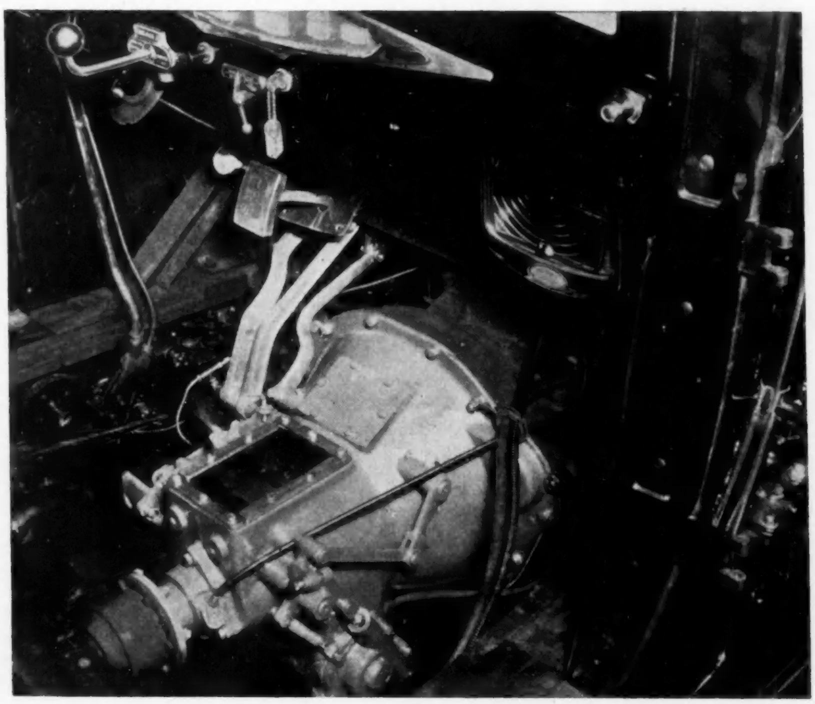

Fig. 21—The Mono-Drive Transmission Installed in a Car
The general dimensions are not appreciably different from those of a conventional transmission. The small pedal at the left of the brake pedal can be depressed at any time to effect an instantaneous shift from high to second speed.

Fig. 22—Centrifugally Acting Clutch of the Mono-Drive Transmission

Fig. 23—Planetary Assembly and Automatic Expanding Spring-Type Clutch Which Is Attached to the Planetary Carrier of the Mono-Drive Transmission

March, 1931, pp. 501-503; *Automotive Industries,* vol. 65, Sept. 12, 1931, pp. 396-400; *The Automobile Engineer,* vol. 21, December, 1931, pp. 596-600; *Machinery,* vol. 37, No. 7, p. 501; and *British Patents,* that is, the classified index of patents on hydraulic presses, meters, motors and the like

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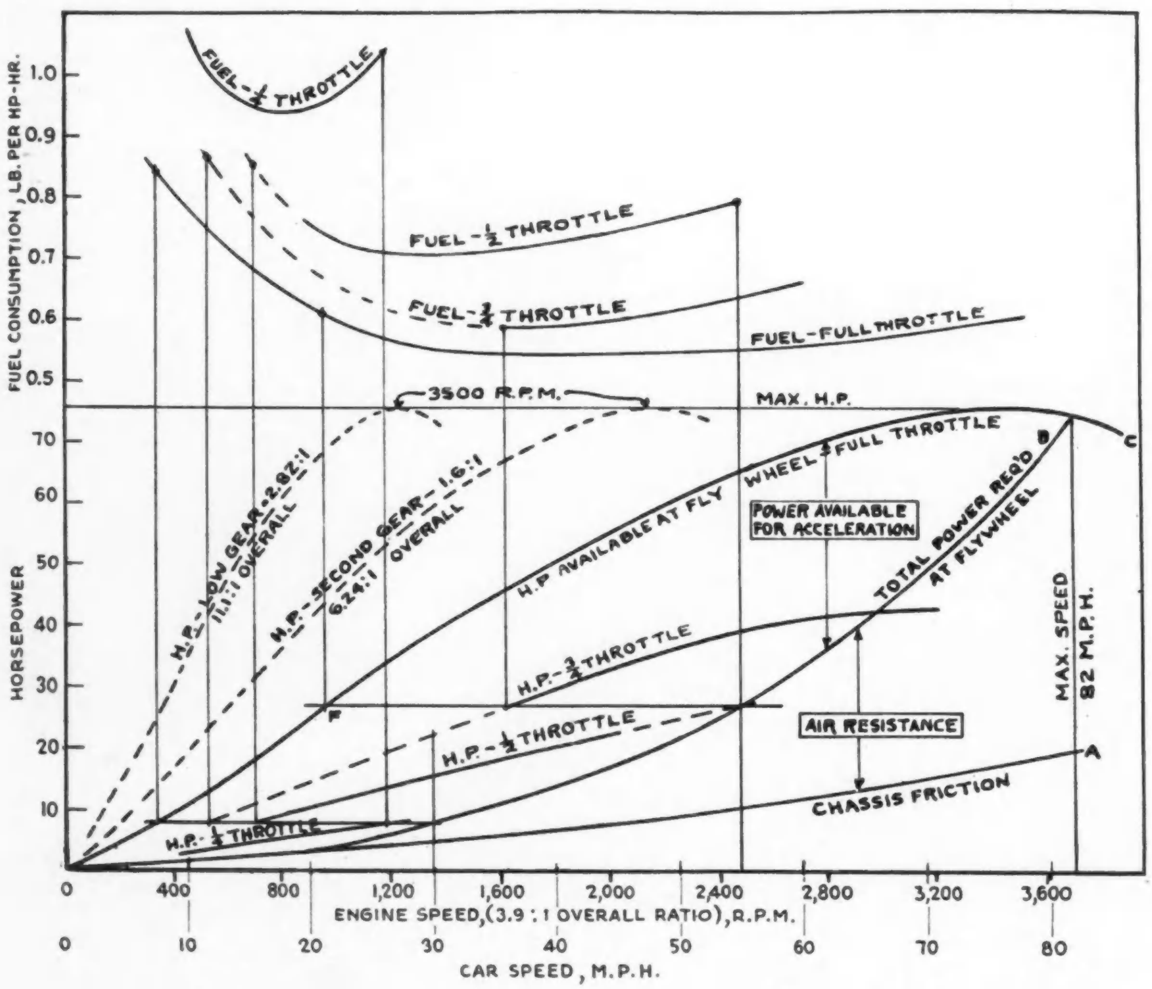

Fig. 25—Relations between Engine Horsepower, Car Horsepower, Car Speed and Fuel Consumption

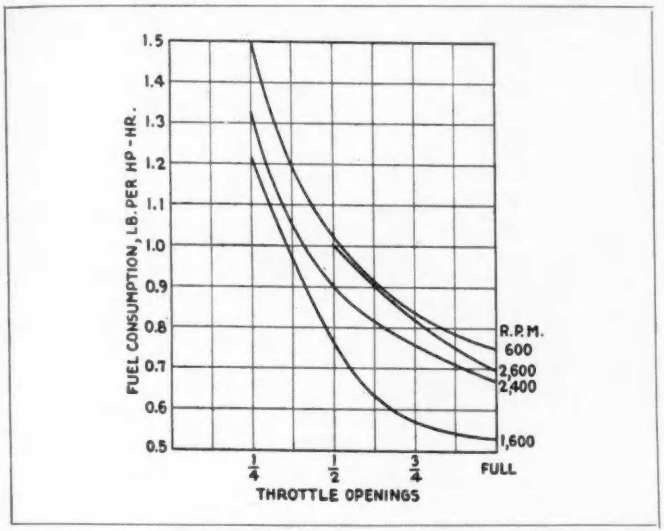

Fig. 24—Fuel Consumption of a Six-Cylinder Engine at Various Engine Speeds and Throttle Openings

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Discussion

Joseph A. Anglada[1]*:*—A good job in compiling a record of the literature descriptive of unconventional forms of motor-vehicle transmissions has been accomplished by Mr. Keys. Descriptions of several other additional attempts to provide transmissions having the characteristics of the simple disc and wheel-type friction transmission have not been published and are therefore not available. Many efforts to provide automatic, or at least semi-automatic, change-gear devices have been made and the defects of complication, unreliability and high cost have been common to all.

The electric form of transmission has had its adherents, but it seems that the penalty of weight and cost prohibits the consideration of electric drive on highway vehicles. Electric-drive systems which have been carefully worked out for bus and truck use weigh upward of 2000 lb. per vehicle and cost about $1 per lb. The flexibility of electric drive is the novel feature desired in any form of motor-vehicle propulsion and the simplicity of control also adds to its attractiveness, but the penalization of weight and cost seems to be insurmountable.

The difficulty of devising a mechanical mechanism possessing the good characteristics of the electric drive has prompted the development of means to make gear changing easier. Several creditable attempts along this line have resulted in the present quite satisfactory syncromesh transmission, power-operated clutch and free-wheeling unit, a triple combination. This combination, however, does not provide a continuous propulsive torque nor a ratio range sufficiently large, but does facilitate manual gearshifting.

To provide a means for automatically obtaining the optimum gear ratio, it seems advisable to be able to change ratios without interrupting the flow of power to the road wheels. This suggests the use of a planetary gear-system with its brake bands and linkages, or the use of over-running clutches with a sliding gear or clutch-type transmission. Full automatic control does not seem advisable, because slippery road and grade conditions frequently make it advisable to anticipate the use of a lower gear; whereas, if the automatic control depended on increase of torque demand to obtain lower gear-

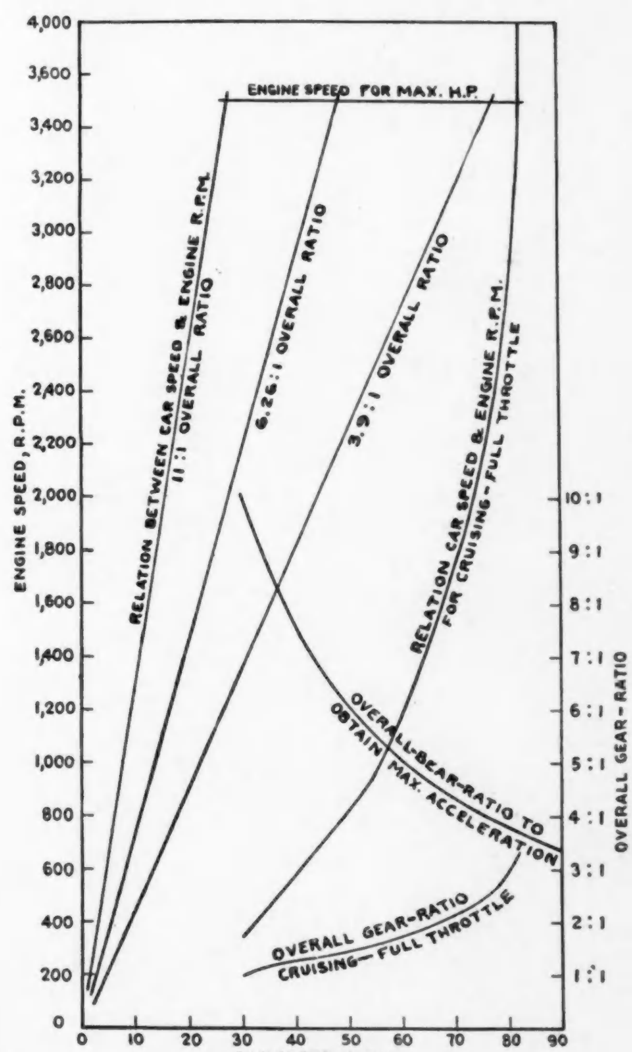

Fig. 26—Relations between Engine Speed, Car Speed and Over-all Gear-Ratios

[1]M.S.A.E.—President, Anglada Motor Corp., New York City.

ratio, the reduction of traction due to a slippery road-surface would produce the opposite effect. This indicates the advisability of providing a manual control to fix the highest gear ratio available under any set of conditions; then, lower gear ratios could be automatically obtained to meet increased torque demands of the driving wheels.

It appears that any substitute for our present generally used transmission system must meet the following specifications. For a passenger car requiring a 70-hp. engine, the combined weight of the engine flywheel, clutch and transmission—or their substitutes—must not exceed 150 lb. and the manufacturing cost of the new mechanism to replace the three conventional units must not exceed $25. Further, the control of the transmission ratios must be more simple and easy than at present, and the torque transmission must be continuous throughout the ratio range of the device. Of course, any

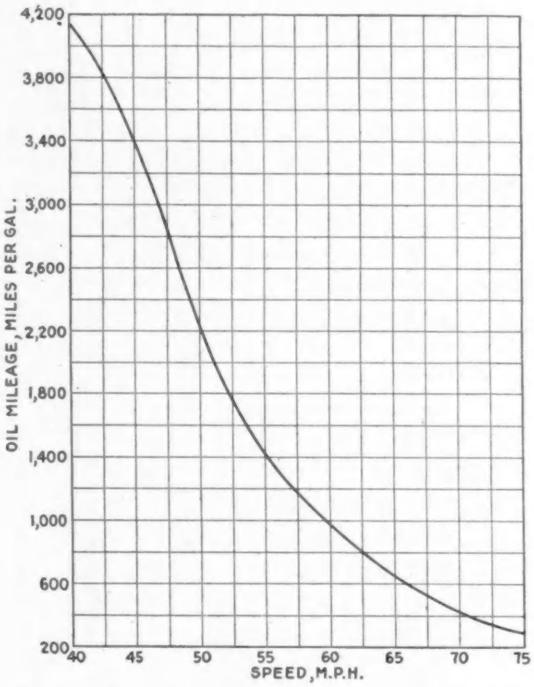

Fig. 27—Oil Consumption in a New V-Type Eight-Cylinder Engine at Various Road Speeds

change of speed ratio must be accompanied by a proportional converse change in torque.

Whether an automatic torque-responsive means for controlling the transmission is advisable, only experience in the hands of the public can determine.

Oscar H. Banker[2]:—It has been stated by Mr. Keys that the automatic transmission can contribute to economy of operation, comfort and speed, but not to safety. In the opinion of hundreds of motor-car operators who have driven the Banker Mono-Drive, safety is one of the greatest contributions which it makes to driving. Mr. Baker, of the National Safety Council, and Mr. Newton, of the Travelers Insurance Co., both of whom have made it their business to study safety from all angles, did not hesitate to put their opinions in writing. They both agree that the elimination of the necessity that the driver must think about the shifting of gears will make driving safer. So long as the operator has to remember any buttons or valves or levers by which

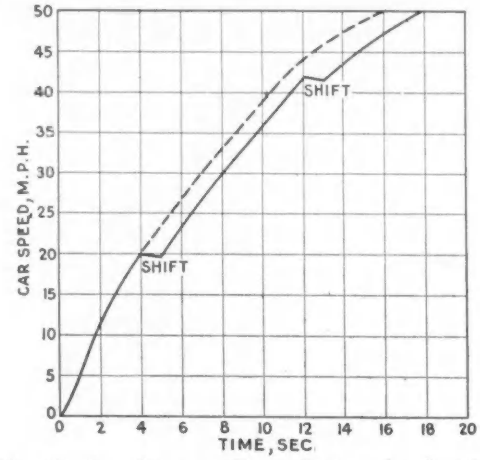

Fig. 28—Speed versus Time, During Acceleration Through Gears in a Modern Car

to accomplish shifting, no matter how simple their operation, it will require thinking and a quick decision by the driver in case of emergency. When using Mono-Drive, no such thinking or decision is necessary for shifting. All the operator has to think about is to avoid the danger by stopping and steering. The engine will never die in such cases; the operator does not need to remove his hands from the steering wheel nor his attention from the road. Mono-Drive is further fortified with additional safety by the No-Rol-Bak, which is absolutely foolproof. All through the development we kept constantly before us and believe we have attained the three features of foolproofness, flexibility and safety.

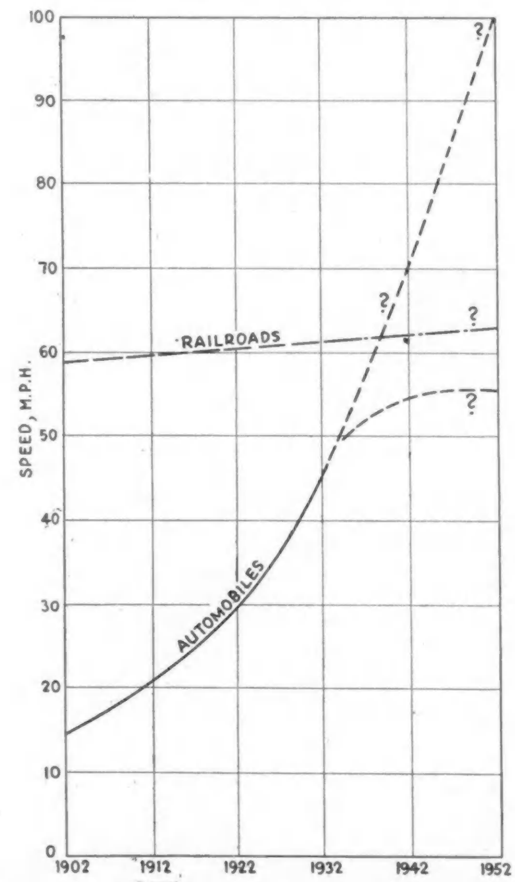

Fig. 29—Average Cruising Speed with Modern Cars, with Forecasts of Future Speeds

[2]Magill-Weinsheimer Co., Chicago.

Commercial Flight Tests Improved by New Equipment and Methods

By A. L. MacClain and D. S. Hersey

THE purpose of this paper is to show that the value of commercial flight-testing depends largely upon the utility, reliability and accuracy of the equipment and instruments employed. In general, it is found that improvements in these three factors depend largely on the simplicity of the apparatus used.

Special engine-tests have led to the development of instruments and apparatus not commonly available commercially, and the use of these has made it possible to employ the aircraft engine as an instrument for the measurement of power in flight. This ability to measure power has led to the development of a method of making comparative engine-temperature and other tests which eliminates or corrects for a number of the major variables ordinarily affecting such tests.

Because of the increasing popularity of the controllable-angle propeller, tests are also described which enable one to determine the value of such a propeller when only an adjustable-angle propeller is available.

IN the early days of aeronautics, the chief object of a flight test was to find out whether the airplane would or would not fly. Now, however, ability to fly is taken for granted, and flight testing has evolved into a somewhat complicated technique for determining the performance of an airplane in terms which are accepted as standard among aeronautical engineers.

For many years, flight testing has been a regularly organized activity of government military services, and more recently it has become an important part of the work of the authorities responsible for the licensing of civil aircraft. The military flight-test organizations, however, can ordinarily deal only with airplanes purchased by the Government, and the tests of the civil authorities must generally be limited to determining the extent to which an airplane meets certain minimum requirements. The manufacturer, therefore, must make his own tests, even if only to insure that his airplane will not fail in tests made by the Government.

We realize that flight testing is by no means a new subject and that in the last few years it has been ably discussed in papers read before the Society and elsewhere in the technical press. We believe, however, that, as commercial units engaged wholly in airplane flight-testing are not very common in the United States, there may be some interest in an account of the equipment and methods employed in such an organization, not only for handling the common problems of flight testing but also in meeting certain special requirements.

Flight testing has been carried on in Hartford, Conn., almost from the time of the first establishment of aeronautical industries there. The object of a large part of the work has always been to determine the performance of aircraft engines in flight under various conditions. This has resulted in unusual attention to the behavior of the engine but, while it is recognized that some of this emphasis on the powerplant may be superfluous in ordinary flight-testing, it is felt that, since powerplant performance is a major factor in airplane performance, something more should be known about the engine than merely the speed at which it runs.

At this point it may be well to note that the equipment and methods to be described are intended almost wholly for determining such quantitative performance characteristics as airplane speed and engine power, and that what might be called qualitative characteristics, such as controllability and maneuverability, have not been given great attention.

Equipment and Personnel

Until the spring of 1930, our flight testing was carried on with very limited personnel and equipment. With the transfer of activities to the new Rentschler Field in East Hartford, however, a considerable increase in facilities became possible, chief among which is a hangar designed expressly for experimental flight-testing. The central part of the building affords a clear floor-space of 125 x 80 ft., with a clearance of 24 ft. under the roof trusses. On one of the 125-ft. sides, at right angles to the edge of the airport, is an electrically operated cantilever door, a piece of equipment which has well justified its cost. On the end of the building away from the field is a one-story lean-to, 80 x 30 ft., containing the stock room, machine shop and metal shop. On the other end is a

[This paper was presented at the 1933 Annual Meeting of the Society.]
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D. S. Hersey is assistant aeronautical engineer, United Aircraft & Transport Corp., East Hartford, Conn.

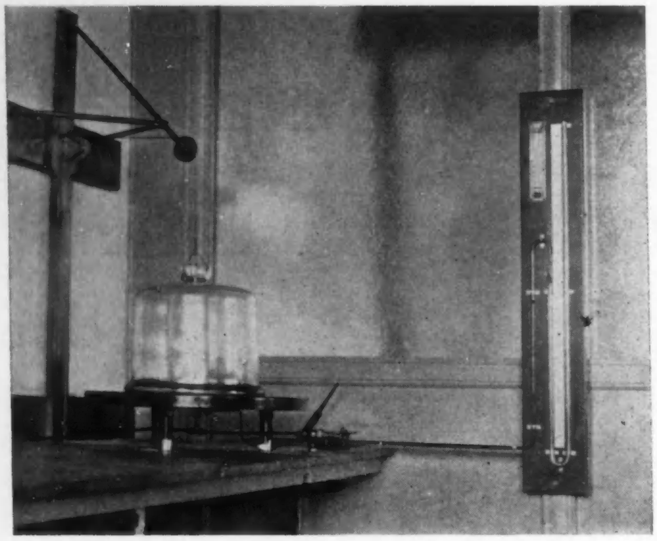


Fig. 1—Bell-Jar Apparatus for the Calibration of Altitude Instruments

two-story lean-to, 80 x 20 ft., on the second floor of which are the offices of the chief test-pilot and the flight-test engineers, together with an instrument room. These offices have windows looking out on the airport, and also into the main hangar-space. The first floor of this lean-to has a room for the storage of tested equipment, as well as an oil storeroom and a locker room.

Sunk in the center of the hangar floor are two sets of Fairbanks scales, arranged so as to form a "T." The larger of these is a hay-scale having a platform, 24 x 3½ ft., flush with the floor and forming the head of the T. This scale, which has a capacity of 40,500 lb., supports the airplane wheels during weighing. The other unit, at right angles to the first, supports the tail skid and consists of a 44-ft. beam, 12 in. wide, mounted on a scale at each end. Each of these end scales has a capacity of 10,200 lb. All three balance-beams are sunk into the floor adjacent to the scales and are covered with hinged floor-plates when not in use. As the beam balance-weights are of the runner type, no handling of loose weights is necessary.

Considerable equipment for constructing experimental cowling and other special parts has been installed in the shop. This includes sheet-metal and welding machinery as well as lathes, drill presses, a grinder, a band saw, benches, vises and the like. The following personnel is employed at this hangar solely to do experimental work: Two engineer test-pilots, one flight-test engineer, five experienced general mechanics including one who can act as an observer, and one sheet-metal worker.

The use of this hangar exclusively for experimental work has proved to be of considerable advantage in that much of the inevitable conflict with other types of flying has been eliminated. Although we do some flight testing of new airplanes, by far the greater part is of new engines and of engine modifications. For this reason, two airplanes are maintained for use as permanent flying laboratories. Of these, one is a high-performance pursuit-type biplane and the other is a medium-performance observation-type biplane.

A great deal of special equipment and a few special features were incorporated in these airplanes. Some are:

(1) Detachable engine-mounts

(2) Permanent thermocouple wiring and special switch-installations for taking cylinder temperatures

(3) Special pressure-gages and temperature gages for measuring engine pressures and temperatures

(4) Oxygen equipment for high-altitude flights

(5) Aluminum tubes from the cockpits to the engine compartment and to the interplane struts so that instrument tubes and wires can be easily installed without affecting drag

(6) Special tanks for measuring fuel consumption

(7) Permanent barograph mounting-lugs

(8) Special, accurate, airspeed indicators, altimeters and tachometers

(9) Manometers for measuring small pressure-differences in air scoops and around cowling parts, as well as for measuring exhaust-manifold back-pressure

(10) Permanent wet-bulb and dry-bulb strut-thermometers for the measurement of air temperature and humidity

(11) Extra pitot-static heads for use with instruments other than airspeed indicators

It was decided to purchase bell-jar equipment for calibrating barographs, altimeters and the like; manometers for calibrating pressure gages and checking airspeed indicators; temperature apparatus for checking thermometers; pyromillivoltmeters, potentiometers, oil-temperature gages, and tachometer calibrating-apparatus. Before the equipment was installed, a few flight tests were made. It was apparent that our results were open to criticism, mainly because our flight-test instruments had not been calibrated. A few very rough checks demonstrated that the results obtained with uncalibrated instruments were unreliable and that complete and good calibrating apparatus was very essential.

The instrument-calibration room in the experimental hangar is 14 x 17 ft., having benches built along three walls and such additional features as electric wall-plugs, a sink with hot and cold running water, an air-pressure outlet and an illuminating-gas outlet. There is a large metal locker for the storage of instruments. It was found that this equipment considerably simplified our work in designing the calibrating apparatus. Inasmuch as this apparatus, although not strictly original, is very simple and easy to use, it is believed that a somewhat detailed description may be of benefit to others who contemplate the purchase of similar equipment.

The bell-jar apparatus is shown in Fig. 1. The bell jar itself is 12 in. in diameter and 10 in. high; it is hung and

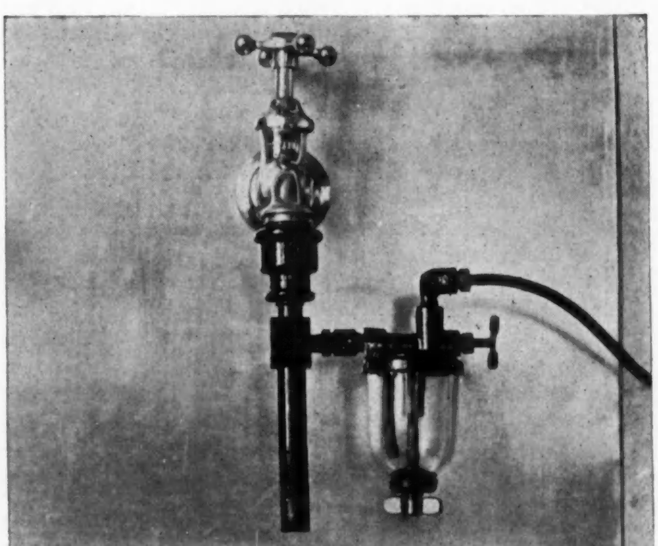


Fig. 2—Water-Aspirator Vacuum-Pump with Water Trap

counterbalanced on an old airplane-control cable. The rest of the apparatus consists of a cast-iron bell-jar base of the design of the Materiel Division, U. S. Army Air Corps, a water-aspirator type of vacuum pump with a suitable water-trap and check valve as shown in Fig. 2, and a mercury barometer of our own design and construction. The water trap is simply a modified automobile gasoline-filter. The mercury barometer is a closed-end "U" tube which is connected directly to the bell jar so that the absolute pressure inside the jar can be measured without any corrections for outside atmospheric pressure. The tube which leads from the bell-jar base to the barometer is restricted so that breakage of a bell jar or any quick change in pressure will not cause the mercury in the barometer to surge and break the closed end of the tube.

The apparatus for calibrating airspeed indicators and pressure gages is shown in Fig. 3. This consists of a water manometer and a mercury manometer connected in parallel to three lines. One of these lines is connected to the compressed-air supply by a paint-gun air-cleaner and reducing-valve assembly. The second line is connected to the aspirator, and the third to the instrument being calibrated. The installation of shut-off cocks in the lines makes it possible to use the mercury column independently of the water column and to obtain pressures both above and below atmospheric.

The temperature apparatus, shown in Fig. 4, is very simple. It consists of two Bunsen burners which are used to heat an oil pot supported on a tripod stand. The temperature "standards" are two thermometers of different ranges. One is used for temperatures above 100 deg. fahr. and has a range of 100 to 800 deg. fahr.; the other is used for temperatures below 100 deg. fahr. and has a range of —70 to +110 deg. fahr. These thermometers are hung on a support above the oil pot during the calibration of temperature instruments.

The design of the tachometer calibrating-apparatus, shown in Fig. 5, was copied from similar apparatus used by United Air Lines, Inc., in Chicago. It consists of a 1/6-hp. constant-speed electric-motor connected by a coupling of rubber hose to a variable-speed transmission of the friction-disc type. This transmission is a commercial article manufactured by a

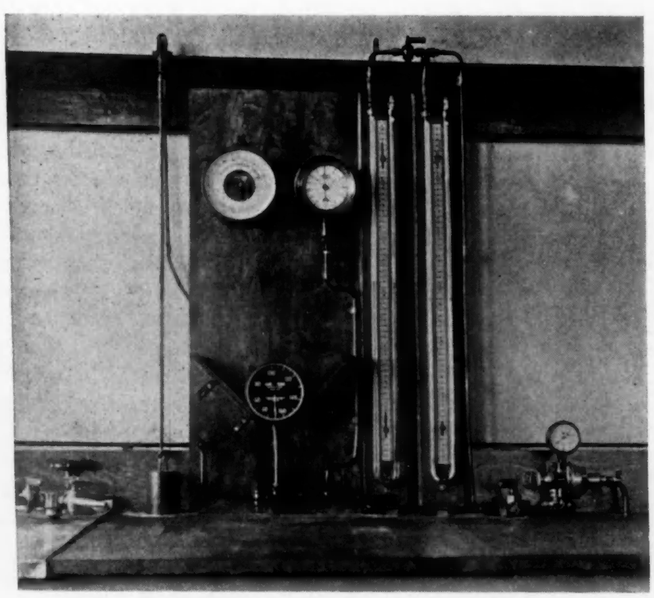


Fig. 3—Apparatus for the Calibration of Airspeed Indicators and Pressure-Vacuum Gages

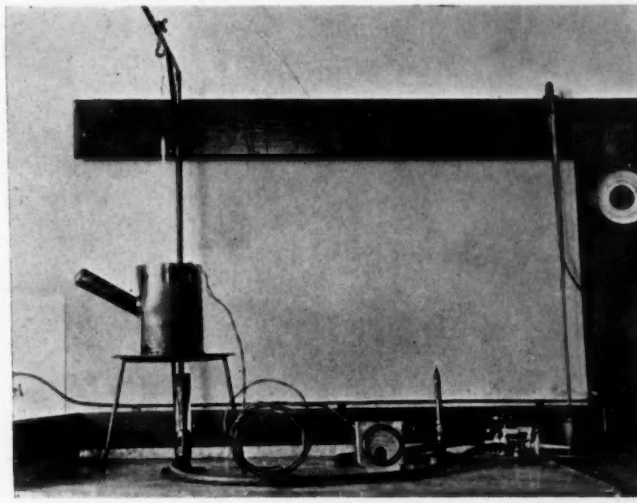


Fig. 4—Apparatus for the Calibration of Temperature-Measuring Instruments

scientific-instrument company. The driven shaft of this variable-speed transmission is connected to a ½ to 1, two-way adapter which in turn drives two flexible shafts. One of these shafts runs a centrifugal-type "standard" tachometer which, by means of an integral revolution-counter, has been carefully calibrated by timing with a stop watch. For ordinary checking, the tachometer which is connected to the second flexible shaft is merely compared with the indications on the standard tachometer. For very accurate calibration, however, the tachometer to be calibrated is checked against timed revolutions as was done for the standard tachometer. This apparatus is accurate to about 5 r.p.m. and therefore is sufficiently good for all flight-testing needs. In fact, most tachometers cannot be read to within 5 r.p.m. Because most apparatus of this sort which is sold commercially is very expensive, it seems worth mentioning that all this calibrating apparatus, exclusive of benches, locker and the like, cost less than $270.

It was formerly our practice to calibrate flight-test instruments just before a flight test, just afterward, or at both times. Experience has shown, however, that certain of the instruments could be counted upon to hold a calibration for a considerable period. Those which generally could thus be relied upon were almost always barographs, tachometers and altimeters. Some of them, especially the barographs, have been known to hold the same calibration for over a year. Because of this reliability, these instruments have lately been calibrated only once every two or three months. The other less-dependable instruments, such as supercharger pressure-gages, airspeed indicators and pyromillivoltmeters, have usually been checked before each new flight-test, and some of them during and after the tests. The current calibrations, which are marked with the type, date, the instrument number and a serial number, are kept in a loose-leaf book; all obsolete calibrations are removed and filed as soon as new ones are inserted, and an index makes it easy to find these old calibration records in case they are needed for further anaylsis of test data.

Periodic calibrations soon showed us that the ordinary commercial airplane-instrument could not be relied upon for use in accurate flight-tests. Supercharger pressure-gages and airspeed indicators could not be counted on to hold their calibrations, tachometers were found to have "up" and "down"

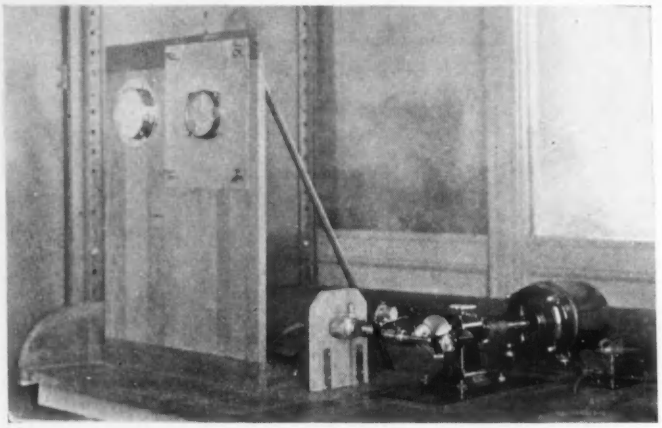


Fig. 5—Tachometer Calibrating-Apparatus

lag of as much as 50 r.p.m., and altimeters were discovered to be far from accurate. Several special recording-instruments were bought to replace the unreliable commercial instruments: A recording tachometer, a recording airspeed-indicator, a strut-temperature recorder and a barograph. See Fig. 6.

For the most part, these recording instruments were not a great success. The recording tachometer did not have a scale wide enough for accurate determination of engine speed, was awkward to install and was subject to frequent failures. The strut-temperature recorder was accurate and reliable and had a scale wide enough to be easily read, but it too was bulky and extremely hard to install. Another drawback was that the sensitive bulb was so large that it was suspected of creating considerable drag. The airspeed recorder was quite good and also was reliable, but did not have a range wide enough for use in high-performance airplanes. For this reason, repairs were periodically necessary, especially after the pilot unthinkingly had come down from a considerable altitude at a high rate of speed. Inasmuch as only one of these recorders was available, it was continually being changed from one airplane to the other, and for each new installation the whole airspeed system had to be recalibrated over a speed course.

The barograph is the best of the four recording instruments. It is of the double-traverse type and therefore has a wide-enough scale. It is reliable, holds a calibration well and is easy to mount in an airplane. Its main defect is that it measures cockpit pressure rather than free-air pressure and therefore does not record climb performance accurately. We have considered mounting the barograph in a sealed case which could be connected to the static side of the airspeed indicator or to an extra static tube mounted on an interplane strut, but have not yet had an opportunity of trying such an installation.

Coordination Difficult

Because of these troubles with recording instruments and the fact that it is difficult to coordinate them with respect to time, the use of all except the barograph has been practically discontinued. Some thought has been given to photographic recorders and also to photographic recording of indicating instruments. Because so many of our tests are of short duration, and also because such apparatus requires a dark room and special developing-equipment, it is felt that the use of any photographic recording method would hardly be justified. The most satisfactory results have been obtained with special indicating-instruments which were held to rigid specifications when purchased. Of course, these special instruments are more expensive than the ordinary commercial type, but we believe that they are worth the extra cost many times over.

One of these instruments which has been extremely useful is the Kollsman sensitive-type altimeter shown in Fig. 7. This instrument is not only an accurate altimeter but also a very sensitive statoscope for making level-flight runs at altitudes. As the scale is relatively open, there is likewise good reason to believe that this altimeter, when used in conjunction with a split-second timer, would give more accurate climb results than the usual barograph. Provision is made on the instrument for connecting it to the static line of the airspeed indicator. This has been found to be very important, as cockpit pressure-variations sometimes introduce an error of as much as 350 to 400 ft. in altimeter readings.

For test purposes, the best type of engine-speed indicator is believed to be a centrifugal tachometer. Several good tachometers of this type are now being made in this country but, for very accurate propeller-setting tests in level flight, we have found that an instrument such as the Reliance tachometer which has an integral revolution-counter is most useful. With this tachometer, it is possible to time the engine speed in flight, with a stop watch, to within about 5 r.p.m.

Airspeed indicators operated by pitot-static heads are used to determine airspeeds at altitudes. These are always calibrated over a speed course in the usual manner. To obtain improved accuracy, special indicators graduated in 2-m.p.h. divisions are used. As some indicators which have the length of their scale divisions reduced at the higher speeds have been found to have irregular calibrations, this type of instrument has been avoided unless the transition point was well above the normal maximum speed of the airplane being tested.

Since the ordinary, cheap, commercial thermometer has a very serious lag at the lower temperatures, is likely to be inaccurate and has too narrow a range, it is inadequate for measuring strut temperatures. Such of these thermometers as were used were therefore replaced with special, large, capillary thermometers of laboratory grade. These have a lag of only a few seconds. By mounting them in fairing strips with large division-lines and figures, their readings were made easy to determine. See Fig. 10. Total-immersion thermometers about 18 to 20 in. long are easy enough to read and a scale range of —70 to +110 deg. fahr. is deemed sufficient for any atmospheric condition which is likely to be encountered.

Three types of stop watches have been tried for timing airplane speeds over a course, engine-revolutions, take-off times and the like. See Fig. 8. The first type was an ordinary split-second timer with 1/5-sec. divisions. For airplanes with speeds of 180 m.p.h. or over, an error of 1/5 sec. represents an error of about 1 m.p.h. when tested over a 2-mile speed-course. To cut down this error, a 1/10-sec. split-sweep-hand stop watch was purchased. This cost about twice as much as the 1/5-sec. timer. It would have been accurate enough except that its construction was such that it could not be depended on for better than 1/5-sec. accuracy. This stop watch is now used mostly for determining take-off times. It is also used occasionally, in conjunction with an altimeter, for timing climbs. The split sweep-hand lends itself very nicely to this kind of timing. Due to the 1/5-sec. accuracy-limit, however, it is no more useful for speed-course timing than is the ordinary 1/5-sec. timer. For this reason, another watch with 1/20-sec. divisions was sought so that an accuracy of at least 1/10 sec. could be had. Only very expensive timers of this kind could be obtained; but, curiously enough, it was found that 1/100-sec. timers could be bought very reasonably. Although

a few practical experiments indicated that the personal error in speed-course timing was about 1/10 sec., one of these timers was purchased to obtain at least 1/10-sec. accuracy in the time measurements. All of the more accurate speed-course runs since that time have been made with this instrument. One ordinarily would believe that such a high-speed timer would have a very short life; but, thus far, it has given much less trouble than has the 1/10-sec. timer.

Since we have discarded most of the ordinary commercial articles and have been using special ones of extra good quality, there has been very little trouble with instruments.

Special Instruments and Apparatus

It has been our practice for some time to record all data which we believe might possibly be pertinent to our testing, even if no immediate use for the data is apparent. In addition to the usual oil temperatures and pressures, engine speed, airspeed, altitude and strut temperature, we now record whenever possible: (*a*) engine intake-manifold pressure, (*b*) temperature of the air entering the carbureter, (*c*) atmospheric wet-bulb and dry-bulb temperatures, (*d*) cylinder temperatures and (*e*) fuel consumption. As these measurements are not ordinarily made by airplane operators, special instruments and apparatus had to be devised either by us or in cooperation with some instrument company. For the measurement of intake-manifold pressures, we first used a sealed altimeter. After a short time, however, limited production on supercharger pressure-gages was started by one manufacturer and has lately been undertaken by another, so that these gages are now obtainable from at least two companies.

When these gages were first made, they were very liable to breakage if the pressures were changed quickly. This happens when an engine throttle is snapped to the closed position. Even with a restriction hole 0.0135 in. in diameter installed in the pressure line to damp hand-vibrations, it was necessary to close a stop cock in the line before pulling the throttle back. Some of the very latest gages apparently have been built with these weaknesses in mind, as they now have built-in restrictions and will stand considerable abuse.

Carbureter-air temperatures are comparatively easy to determine by means of a thermocouple junction attached to the screen just below the carbureter. Temperatures are measured by the pyromillivoltmeter or potentiometer, which is used to measure engine-cylinder temperatures. In this connection, it might be well to mention that both the pyromillivoltmeter and the potentiometer (Fig. 9) which we use for measuring these carbureter-air temperatures and engine-cylinder temperatures were developed by the Lewis Engineering Co. in close cooperation with our organization. When a pyromillivoltmeter or potentiometer is not available, the bulb of a distant-reading thermometer, rather than the thermocouple junction, is installed below the carbureter. As this type of instrument is likely to be unreliable at low temperatures, however, it is not used unless it is absolutely necessary.

Atmospheric humidity can be determined simply by using a wet-bulb strut-thermometer in conjunction with the regular dry-bulb thermometer. Another psychrometer, of an electrical type, was developed for us by the Lewis Engineering Co. This consists of a pyromillivoltmeter, calibrated from 0 to 50 deg. fahr., which is connected to a thermopile. The "cold junction" of this thermopile is enveloped by an exposed wick which is led to a water container. The "hot junction" of the thermopile is merely exposed to the atmosphere. The thermopile and water-well unit is mounted on an interplane strut and wire extensions are run to the pyromillivoltmeter mounted in the airplane cockpit. The instrument indicates the difference between the wet-bulb and the dry-bulb temperatures, and the dry-bulb temperature is determined from the strut thermometer. Both types of instrument have proved quite satisfactory, although the electric type is somewhat the more sensitive. Both are shown in Fig. 10. Vapor pressures are determined by use of a specially prepared chart.

For the measurement of fuel consumption, two types of fuel meters have been developed for use in conjunction with stop watches and are shown in Fig. 11. The longer type has

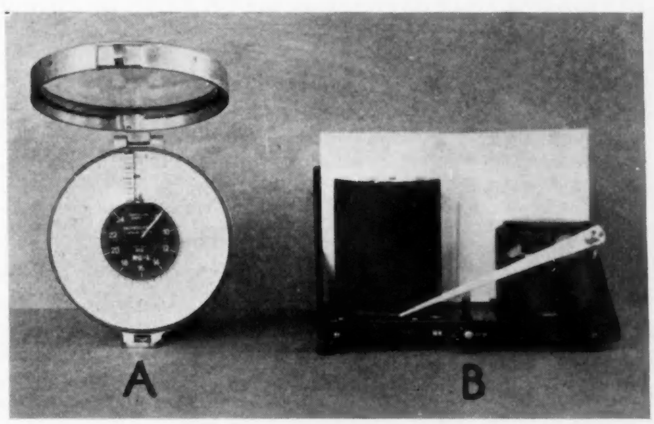
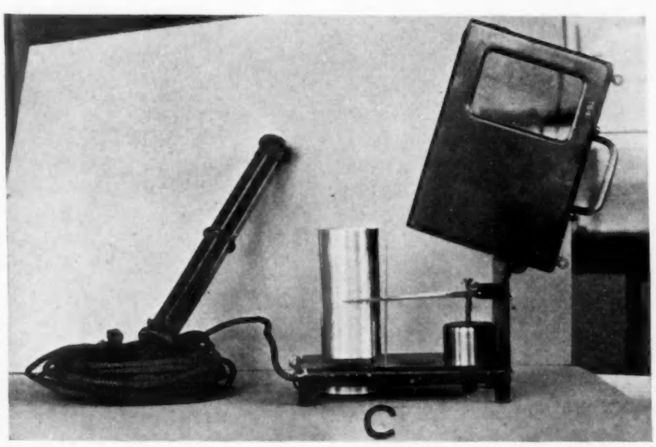
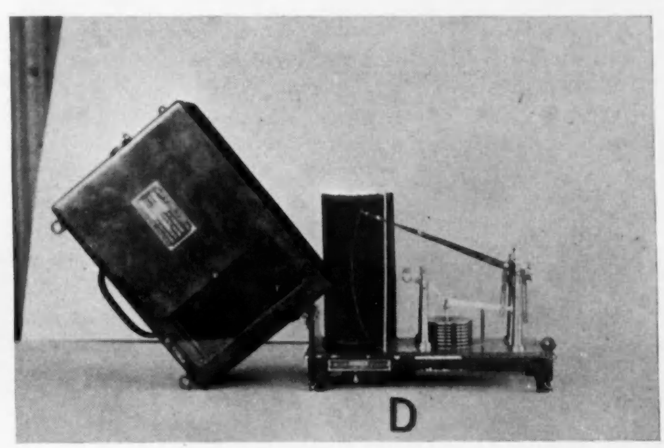


Fig. 6—Recording Instruments Used in Flight Testing

Recording tachometer, (*A*); airspeed recorder, (*B*); strut-temperature recorder, (*C*); and double-traverse barograph equipped with a drum having ½-hr. operating capacity, (*D*)





proved to be no more accurate than the short type and is, besides, somewhat harder to install. Fuel consumption is measured with this apparatus in much the same way as it is on engine test-stands. The brake horsepower during these runs is determined by a method referred to later in this paper.

Several other simple pieces of apparatus of interest are illustrated in Fig. 12. On the left is a propeller protractor which is intended for use in measuring blade angles without removing the propeller from the airplane. This is a development of a type devised by the National Advisory Committee for Aeronautics and was designed by our observer, D. S. Pierce. On the right is apparatus for the measurement of take-off distance and time. Distance is measured by counting the revolutions of a bicycle wheel, and time is measured by a stop watch. Wind velocity is measured with a hand anemometer and stop watch, and the numbered markers are used to spot the point of take-off of the airplane until distance measurements can be made.

Attempts to obtain additional test data, particularly on the powerplant, have led to the development of methods in which the disturbing effects of changes in atmospheric conditions and engine power would be minimized. A method of determining engine power in flight will first be dealt with.

From equations given in Reports Nos. 295 and 426 of the National Advisory Committee for Aeronautics, it can be shown that the brake horsepower of conventional airplane engines varies according to the equation

$$\frac{B.hp.}{B.hp._0}=\left(\left[\frac{P-p}{P_0-p_0}\right]\sqrt{\frac{T_0}{T}}\right)\left(1+\left[\frac{\lambda-\lambda\eta}{\eta}\right]\right)-\left(\frac{\lambda-\lambda\eta}{\eta}\right) \qquad (1)$$

where

$B.hp.$ = Brake horsepower
P = Absolute pressure of the air entering the carbureter, in inches of mercury
p = Water vapor pressure in the air entering the carbureter, in inches of mercury
T = Absolute temperature of the air entering the carbureter
λ = Ratio of mechanical friction to the friction horsepower at sea level
η = Mechanical efficiency of the engine at sea level

Note that the symbols having a subscript $_0$ denote a standard condition, and that those having no subscript denote an observed condition.

Test data indicate that in engines with gear-driven centrifugal-superchargers located between the carbureter and the engine, at constant engine-speed and fixed throttle-setting the absolute pressure in the intake manifold varies directly as the absolute pressure of the air entering the carbureter. This relation should, of course, be corrected for the effect of vapor pressure. Although not strictly accurate, it is convenient and probably sufficiently close to correct both the intake-manifold pressure and the carbureter-air pressure by subtracting from them the vapor pressure of the air entering the carbureter. Thus, the relation between manifold pressure and carbureter pressure is expressed by the equation

$$\frac{P-p}{P_0-p_0}=\frac{P'-p}{P'_0-p_0} \qquad (2)$$

in which

P = Absolute pressure in the intake manifold, and the subscript $_0$ denotes this pressure when the engine is operated at rated speed and power at standard sea-level.

Combining Equations (1) and (2) we obtain

$$\frac{B.hp.}{B.hp._0}=\left(\left[\frac{P-p}{P_0-p_0}\right]\sqrt{\frac{T_0}{T}}\right)\left(1+\left[\frac{\lambda-\lambda\eta}{\eta}\right]\right)-\left(\frac{\lambda-\lambda\eta}{\eta}\right) \qquad (3)$$

Equation (3), which takes account of mechanical friction and friction horsepower, is somewhat inconvenient. With the comparatively small changes in brake horsepower which are involved in ordinary corrections to standard sea-level, it has therefore become common practice to neglect the terms representing friction. This, in effect, involves the assumption that brake horsepower varies in the same manner as indicated horsepower. Even in the complete Equation (3), no allowance is made for the effect of changes in volumetric efficiency due to variations in cylinder temperature and to other causes. The magnitudes of these effects are believed to be small enough so that over a narrow range of power they may be neglected, especially in view of the complication which would result from considering them. This approximation has been made in the remainder of this section of the paper.

Under these assumptions and disregarding the effect of exhaust back-pressure, it is possible to determine engine power in terms of intake-manifold pressure, carbureter-air temperature and atmospheric vapor-pressure, so that an airplane engine can be employed to measure power after being calibrated on a dynamometer.

An engine calibration is made on the dynamometer test-stands of The Pratt & Whitney Aircraft Co. in the following manner:

(1) A convenient engine speed is chosen and the engine is kept running at this speed, at best-power mixture-setting, regardless of the throttle opening. This is accomplished by appropriate variation of the dynamometer torque.

(2) The throttle position is varied in several steps from wide open to about one-fifth open.

(3) For each throttle opening, the absolute intake-manifold pressure, the temperature of the air entering the carbureter, the atmospheric vapor-pressure, and the brake horsepower are determined.

(4) The observed brake horsepower for each throttle setting is corrected to standard carbureter air-temperature (60 deg. fahr.) by the factor

Fig. 7—Kollsman Sensitive-Type Altimeter

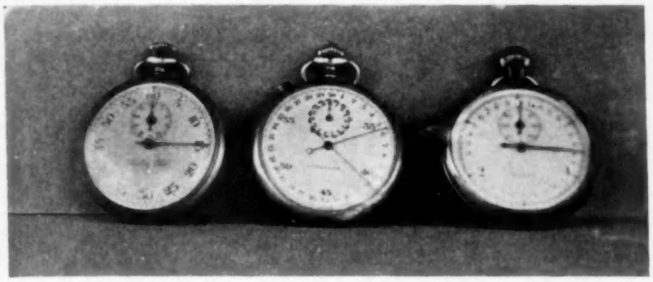


Fig. 8—Three Types of Timers

The instrument at the left measures 1/5 sec.; the one in the center, 1/10 sec. and is equipped with a split sweep-hand; and that at the right measures 1/100 sec.

$$\sqrt{\frac{T}{T_0}} = \sqrt{\left(\frac{459 + t}{459 + 60}\right)} \qquad (4)$$

which is derived from the relation of engine power and air temperature, Equation (1), and in which t is the temperature in degrees fahrenheit of the air entering the carbureter.

(5) This corrected value of brake horsepower is plotted against the value of absolute intake-manifold pressure minus the atmospheric vapor-pressure $(P - p)$ determined for the throttle opening in question.

(6) This same procedure is carried out at several engine speeds and the final engine calibration appears as in Fig. 13.

Back-pressure corrections are ordinarily disregarded during these calibrations, as the back pressure in an engine run on the dynamometer stands of The Pratt & Whitney Aircraft Co. is very nearly constant and corresponds closely to that at standard sea-level.

A study of a considerable number of these calibrations on Pratt & Whitney engines has shown that the power at constant engine-speed varies as follows:

$$\frac{B.hp.}{B.hp._0} = \left(\frac{(P - p) - k}{(P_0 - p_0) - k}\right)\sqrt{\frac{T_0}{T}} \qquad (5)$$

where the symbols have the same meaning as before. The constant k is in the same units as P and p and, from the data thus far obtained, appears to vary from a maximum value of 10 to a minimum of 4 inches of mercury with different engines and at different engine-speeds. It is felt desirable at present to determine the value of k separately for each engine condition. However, a fair average value for Pratt & Whitney engines is $k = 6.5$ inches of mercury.

In flight, engine speed, intake-manifold pressure, wet-bulb and dry-bulb air-temperature, carbureter-air temperature and pressure altitude, are observed. After corrections for instrument error have been made, the brake horsepower is found from the calibration curves by the following procedure.

(1) Determine the atmospheric vapor-pressure from a vapor-pressure chart

(2) Subtract the vapor pressure from the corrected intake-manifold pressure

(3) From the engine-calibration chart, (Fig. 13), determine the brake horsepower which would be obtained at 60 deg. fahr. and which corresponds to the values of "intake-manifold pressure minus vapor pressure" and engine speed obtained in flight

(4) Correct this "brake horsepower at 60 deg. fahr." to the actual temperature-conditions by multiplying by the factor

$$\sqrt{\frac{T_0}{T}} = \sqrt{\left(\frac{459 + 60}{459 + t}\right)}$$

which, it will be noted, is the reciprocal of Equation (4)

(5) This value is corrected for back pressure, taking care to employ the same method of correction throughout any one series of tests.

These engine calibrations are used in flight testing in two ways: first, the engine brake-horsepower can be ascertained very closely at any time during flight if intake-manifold pressure, carbureter-air temperature, atmospheric vapor-pressure and exhaust back-pressure are known; second, the engine brake-horsepower can be regulated in flight to any convenient value if the proper data are previously prepared. Both applications are described in the paper[1] entitled Airplane Flight-Testing for Maximum Speed, by F. M. Thomas and H. W. Fairchild. They are available also in comparative runs for investigating variations in engine temperature with changes in engine cowling. Even if no engine calibration is at hand, the relation given in Equation (5) can be used to obtain comparative results.

Comparative Engine-Temperature Tests

During engine-temperature tests it would be very desirable if the airplane could be flown in an atmosphere of constant density, constant vapor-pressure and constant temperature, and if the airplane speed and the engine power could be kept constant. Inasmuch as engine power depends upon atmospheric conditions, which are constantly changing, all these requirements obviously cannot be met. Nevertheless, it is possible to fly in air of constant density, at nearly constant engine-power and at constant airplane-speed if no changes are made in the drag condition of the airplane between tests. This leaves vapor pressure and atmospheric temperature as variables.

It is relatively simple to fly in air of constant density. The necessary procedure is given in the Thomas and Fairchild paper referred to. It consists essentially in flying, under the atmospheric conditions of a particular day, at the altitude at which pressure and temperature are such as to combine to give the air the desired density.

Since the cylinder temperatures of air-cooled engines are usually most critical at full throttle, it would be desirable to conduct comparative engine-temperature flight-tests at this throttle setting. However, as it is necessary, in remaining in air of constant density, to fly at different atmospheric pressures

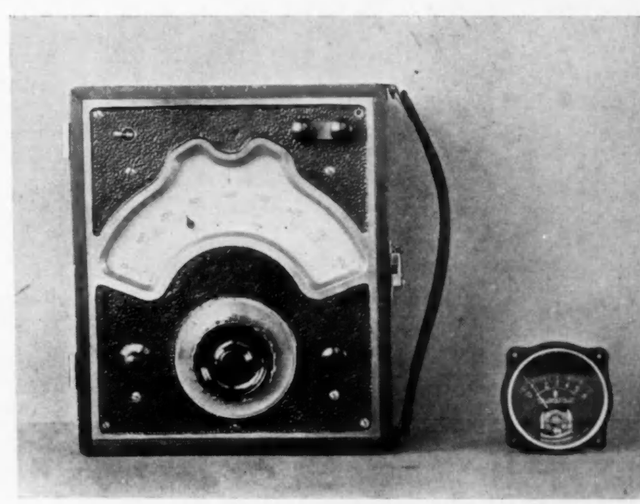


Fig. 9—Instruments for Taking Engine Temperatures

A potentiometer is shown at the left, and a pyromillivoltmeter at the right

[1] See S.A.E. Journal, March, 1933, p. 78.

to compensate for changes in atmospheric temperature, it is apparent that constant power will not be obtained at full throttle and that it is essential to fly at some throttle opening less than the maximum so as to permit the throttle to be regulated in such a way as to compensate for variations in atmospheric pressure and temperature. This is reasonable, as the tests are comparative only.

Suppose that a standard-density altitude of 2000 ft. is chosen as convenient for making the engine-temperature runs, and that the expected variation in strut temperature is 20 deg. fahr. which requires a corresponding change in pressure altitude for the maintenance of constant density. Suppose also that a carbureter-air temperature-variation of 50 deg. fahr. can be expected. Because of these variations the engine power is likely to change by as much as 4 per cent due to changes in atmospheric pressure, and by 4½ per cent due to changes in carbureter-air temperature. The total variation in engine power may be as much as 9 per cent, and the maximum constant power that can be obtained at all times is not more than 91 per cent of the maximum full-throttle horsepower that could be obtained at a 2000-ft. standard-altitude.

Under these conditions it becomes desirable to make the comparative engine-temperature tests, not at full power obtainable at a 2000-ft. standard altitude, but at 91 per cent of it. As temperature changes have practically no effect upon intake-manifold pressure for constant engine-speed, the maximum intake-manifold pressure corresponding to maximum engine power at a 2000-ft.-standard altitude can be found by making a full-throttle run at rated engine speed at a 2000-ft.-*pressure* altitude, and noting the reading of the intake-manifold pressure-gage. This value will be denoted as P_m.

Now if it is assumed that the carbureter-air temperature is at its standard value, and that vapor-pressure changes can be neglected as being of negligible effect, it is apparent from Equation (5) that engine power will vary as $(P - k)$ and that 91 per cent of maximum power at a 2000-ft.-standard altitude will be obtained when the intake-manifold pressure is reduced to a value obtained by the equation

$$P'' = 0.91\ (P_m - k) + k \qquad (6)$$

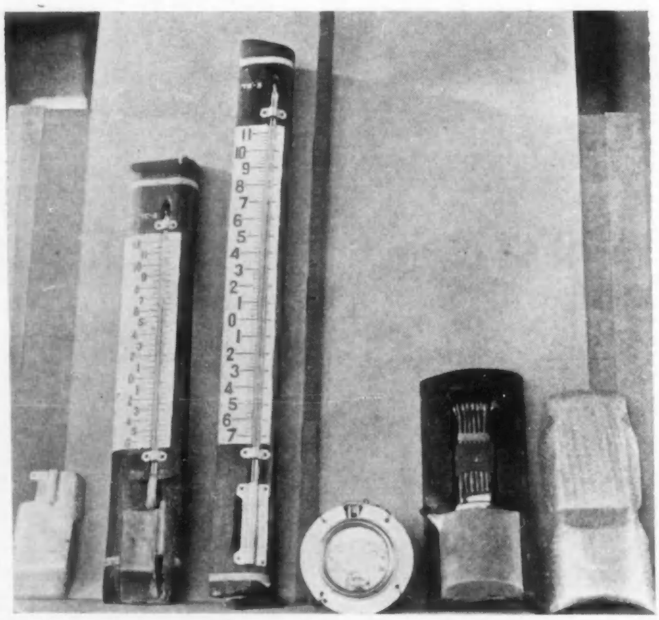


Fig. 10—Psychrometers Used on Airplanes

At the left is a wet-bulb strut-thermometer; in the center, a dry-bulb strut-thermometer; and at the right, a Lewis electric-type psychrometer

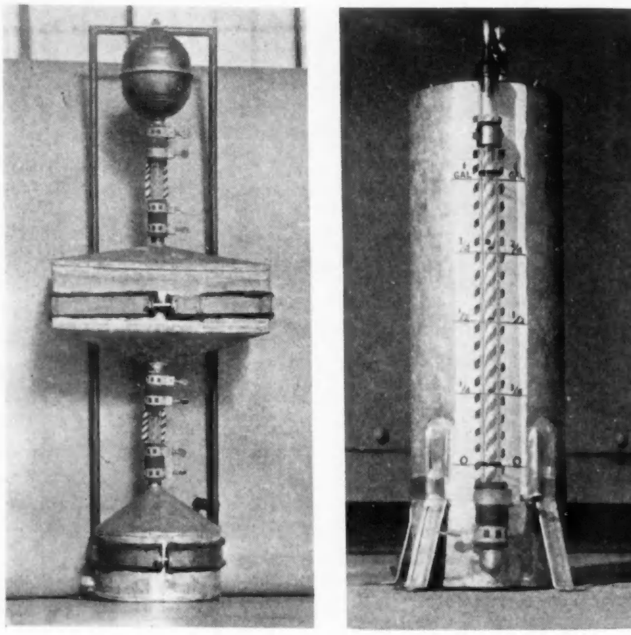


Fig. 11—Fuel-Flow Meters Used in Flight

where P'' is the intake-manifold pressure corresponding to 91 per cent of maximum power at a 2000-ft.-standard altitude. This is true only at rated engine-speed and at a standard carbureter-air temperature. Of course the carbureter-air temperature will not remain constant; so, if constant power is to be maintained, it is necessary to vary the intake-manifold pressure to correct for changes in power which would result from changes in carbureter-air temperature.

If vapor pressure is neglected because of its slight effect, the brake-horsepower correction-factor at constant engine-speed becomes

$$\frac{B.hp.}{B.hp._0} = \left(\frac{P - k}{P_0 - k}\right)\sqrt{\frac{T_0}{T}} \qquad (7)$$

Suppose now that $B.hp._0$, instead of representing the maximum power at standard sea-level conditions, represents the power at which the tests are to be run, then P_0 is equivalent to the value P'' and T_0 is the absolute carbureter intake-air temperature at a 2000-ft.-standard altitude.

As the power is to be maintained at constant value, the ratio $(B.hp./B.hp._0) = 1$. Now if this relation and the previously determined values P_0 $(= P'')$ and T_0 are substituted in Equation (7), the equation becomes

$$1 = \left(\frac{P - k}{\sqrt{T}}\right) \quad \left(\frac{\sqrt{T_0}}{P'' - k}\right) \qquad (8)$$

and $P = C\sqrt{(T + k)}$, where $C = (P'' - k)/\sqrt{T_0}$; or
$$P = C\sqrt{(459 + t + k)}.$$

It is necessary, if a substantially constant power at a 2000-ft.-density altitude is to be obtained, that this relation be plotted for use in flight in some such manner as shown in Fig. 14. This curve is used in the following manner:

(1) The standard density chosen is found at some altitude depending upon the atmospheric temperature

(2) The airplane is flown at this density altitude and at nearly full engine-power until the carbureter-air temperature has had a chance to level out

(3) The carbureter-air temperature is determined and an intake-manifold pressure corresponding to that carbureter-air temperature is read from the curve in Fig. 14

(4) The throttle position is varied until the value of intake pressure read from the curve is indicated on the intake-manifold pressure-gage

(5) The engine speed is checked. If it is found to be different from the rated maximum value, the propeller is re-set to obtain that value at the throttle setting which is necessary to correct the manifold pressure for carbureter-air temperature.

Once this propeller setting is obtained, constant horsepower can automatically be maintained simply by setting the throttle to get the rated engine-speed at the density altitude chosen. This relation holds, however, only as long as the drag condition of the airplane is unchanged. The reason for this can easily be seen if it is remembered that the power absorbed by a propeller of fixed geometric form is dependent mainly upon the airplane speed, the engine speed and the air density. If, however, the airplane drag-characteristics are changed during the series of tests, then the propeller must be re-set to absorb the desired constant power at the different airplane speed.

The method of making engine-temperature runs as described so far has taken account of air density and engine power; changes in atmospheric water-vapor pressure have been disregarded as being small. Changes in power due to changes in back pressure have been neglected for two reasons; first, the changes are likely to be small enough to be negligible in their effect on power and, second, change in back pressure is likely to have very little effect on the cylinder temperatures of an engine. Only strut temperature and airplane speed are left to be dealt with.

Strut-Temperature Changes

As strut-temperature changes are, of course, impossible to regulate, if constant-density altitude is maintained, the next best procedure is to correct for them. It usually is possible to make two or three engine-temperature runs under different conditions of strut temperature, but at constant airplane-speed, constant engine-power and speed, and constant-density altitude. If this is done over a sufficient range of strut temperatures, with all other airplane and cooling factors kept constant, the trend of engine-cylinder temperatures can be plotted against strut temperature. By means of the curves showing these trends, the results of tests of various cooling systems can be corrected to some desired standard strut-temperature. Although no very extensive tests have been made on the variation of cylinder temperatures with strut temperature, it appears from our very incomplete data that, at constant engine-power, constant airplane-speed, and in air of constant density, the cylinder temperatures may possibly vary as the ratios of the absolute temperatures.

The method of making engine-temperature runs just described is used solely for comparative purposes. When the best cowling or baffle condition is found, it naturally is necessary to re-set the propeller for full rated engine-power and then to make a test for maximum cylinder-temperatures in level flight and in climb at "best-power" mixture-setting. It should be remembered that mixture setting plays a very important part in engine-cylinder temperature and that all comparative runs should be made at "full-rich" setting; or, if not at full-rich, then at a fixed-mixture control-setting. The former is ordinarily the simpler and the more convenient in practice, especially as the full-rich setting is less affected than any other by changes in control linkages.

When supercharged engines are used in comparative engine-temperature tests, below critical altitude, the intake-manifold pressure P_0 to be substituted in Equation (8) is taken as that specified by the engine manufacturer for normal operation at the test altitude. Thus, the tests are automatically made at maximum allowable engine power and the only check test necessary at the optimum condition of cooling is one at best-power mixture-setting.

Perhaps it should be mentioned again, in concluding this discussion of engine-temperature tests, that the procedure described does not require a dynamometer calibration of the engine if a good average value of the constant k is known. If, however, the results of such a calibration are available, considerably better accuracy in regulating engine horsepower can be obtained by use of the calibration curve.

It is often desirable to determine the take-off characteristics of an airplane and especially the effect on the take-off char-

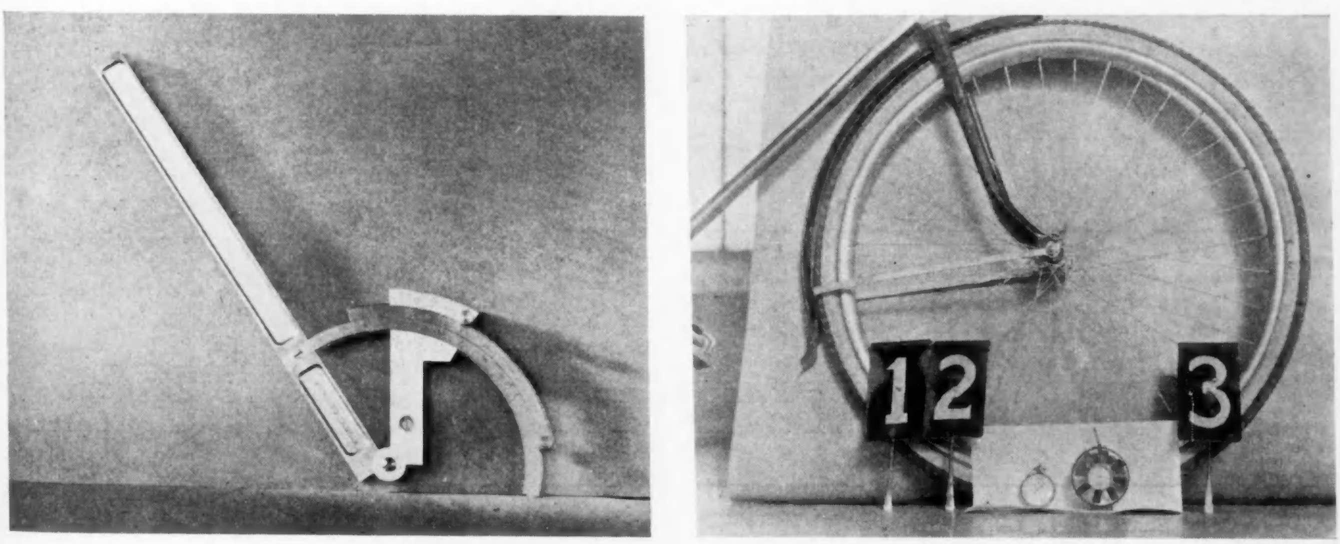


Fig. 12—Take-Off Measuring-Apparatus

A propeller protractor is shown at the left. At the right is a bicycle wheel equipped with a revolution counter attached to its hub. In front of the wheel a stop watch is shown at the left and a hand anemometer at the right. The numerals shown are on markers of the type used for spotting the take-off point.

acteristics of modifications in the powerplant and in the airplane itself. A special technique, some of which may be new, has been developed for making such tests. Since, under most conditions, take-off time, although quite easy to measure, is of relatively minor importance, we are accustomed to give the major attention to the measurement of take-off distance. Tests have been made both with brakes set at the beginning of take-off and with brakes released; but, although it is recognized that the latter condition is the more nearly representative of ordinary flying, our experience indicates that test results are more nearly uniform if the brakes are kept set until the engine has reached its maximum speed with throttle at the allowable opening. In our work, therefore, this procedure is followed, the airplane being started each time from a point marked on the field by a flag or stake.

Several observers stationed near the expected point of take-off note the spot at which the wheels actually leave the ground and each of them walks immediately to this point as he observed it. We find ordinarily that there is little variation in the point of observed take-off among the various observers and, as soon as they are agreed upon it, a numbered metal marker such as is shown in Fig. 12 is stuck in the ground to preserve the point for future measurements. Ordinarily, three take-offs are made under each set of conditions, each point of take-off being temporarily spotted as just mentioned. Upon the completion of the series, one of the observers rides a bicycle from the starting point to the different marked points where the airplane left the ground, and the take-off distance is thus quickly determined from the revolution counter on the bicycle wheel as previously mentioned. If the bicycle has been carefully calibrated for the particular field surface from which the take-offs are made, the distance as thus measured is ordinarily correct within about 2 ft.

The barometric pressure and the air temperature are, of course, observed and recorded during take-off tests and the wind velocity is measured during each individual take-off run by means of the hand anemometer and stop watch previously described. The time of each take-off run is likewise taken with a stop watch and both the static engine-speed and the engine speed at the end of take-off are noted.

It obviously is necessary to correct the observed take-off distances to the condition of "no wind." This can be done conveniently by a method[2] which assumes that, if the wind velocity were equal to the stalling speed of the airplane, then the take-off distance would be zero. Other methods proposed for making this correction give somewhat different corrections, and in order that the results of our tests may not be too greatly dependent on corrections which are possibly doubtful, we make take-off tests only during periods when the wind is reasonably steady and has a velocity not over 8 m.p.h.

[2] See Engineering Dynamics, by Walter S. Diehl; The Ronald Press Co., 1928, p. 227.

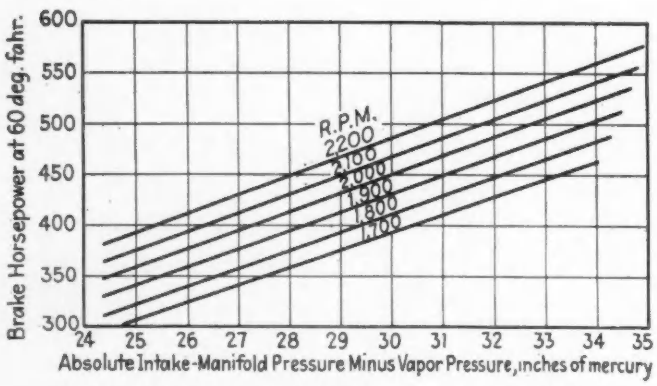


Fig. 13—Typical Pratt & Whitney Co. Engine-Calibration Curves

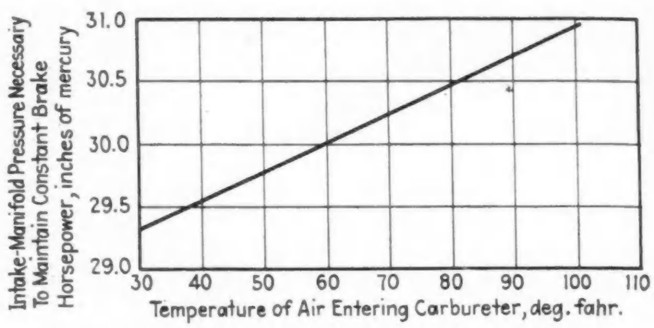


Fig. 14—Typical Curve for the Maintenance of Constant Brake-Horsepower in Flight

The reduction of the results of take-off tests to standard sea-level is a matter of some difficulty, although, as the altitude of Rentschler Field is less than 50 ft., the departures of actual atmospheric conditions at the field from those of standard sea-level are not very great in ordinary weather. Whenever possible, we prefer to make take-off tests during two periods of somewhat different atmospheric conditions, without making any change in the airplane or its powerplant. By plotting the results of these two tests against the observed density altitudes, a line can be drawn to show the general trend of the effect of density altitude on take-off performance. An interpolation then gives the expected take-off characteristics at standard sea-level. Because of the low altitude of the field, it often happens that at least one of the series of take-off tests can be made at an air density greater than that of standard sea-level.

Controllable-Propeller Tests

The interest which recently has developed in the use of the controllable propeller for improving take-off performance often makes special take-off tests desirable. Even though in many cases a controllable propeller is not available, it is nevertheless possible to make simulated tests which give quite reliable results. In such tests, runs are made with an ordinary adjustable propeller, the blades being set successively at different angles so as to give different engine speeds in take-off. The take-off performances obtained in such tests are reduced to standard conditions as already described and are plotted against blade angle as shown in Fig. 15, in which the minimum points of the curves obviously indicate the blade angle for best take-off. During the short flight which follows each take-off, the pilot naturally must limit the throttle opening to one which does not cause the engine to overspeed, for, obviously, the propeller-blade angle which gives an improved take-off may be so low as to permit the engine to turn too fast in level flight at full throttle. In some cases it happens that the best propeller-blade angle, as determined from the curves, is so low as to allow the engine to overspeed even in take-off. In this event, the lowest blade-angle which can be recommended is that with which the engine just reaches its rated speed at the end of the take-off run, a condition that requires reduced throttle opening in climb.

A method analogous to that just described is employed in making tests to investigate the effect of a controllable propeller on airplane climb. Several climbs from about 1000 to

4000-ft. altitude are made at various propeller-blade angles. In each case, the airplane is climbed at full throttle or throttle stop and at such an airspeed that the engine turns at its maximum rated speed. It is desirable to make all the climbs during one day so as to avoid the uncertainties which inevitably arise if it is necessary to reduce the results to standard atmosphere. When atmospheric conditions during the test do not vary appreciably, the climbs can be compared directly on a pressure-rate basis. The results of the climbs are plotted as shown in Fig. 16, in which the highest point on the curve evidently shows the propeller setting for best rate of climb.

A further refinement in the method would consist in running a sawtooth climb at each propeller setting so as to take account of the possibility that an airspeed at which the engine speed was less than rated might give a still better rate of climb.

During tests such as these, it is necessary to watch the engine temperatures very carefully, for there is always the possibility that some combination of conditions may cause these temperatures to go too high.

In conclusion, we recognize that the flight-test methods which we have described are possibly somewhat more complicated, and therefore more expensive, than those which have been customary in commercial work. To a considerable extent, however, we have been forced to these methods because of the inadequacy of those with which we began, and our experience has convinced us that the price which has to be paid for more complete and more reliable data is fully justified by the value of the results.

[1] Assistant professor of engineering research, Pennsylvania State College, State College, Pa.

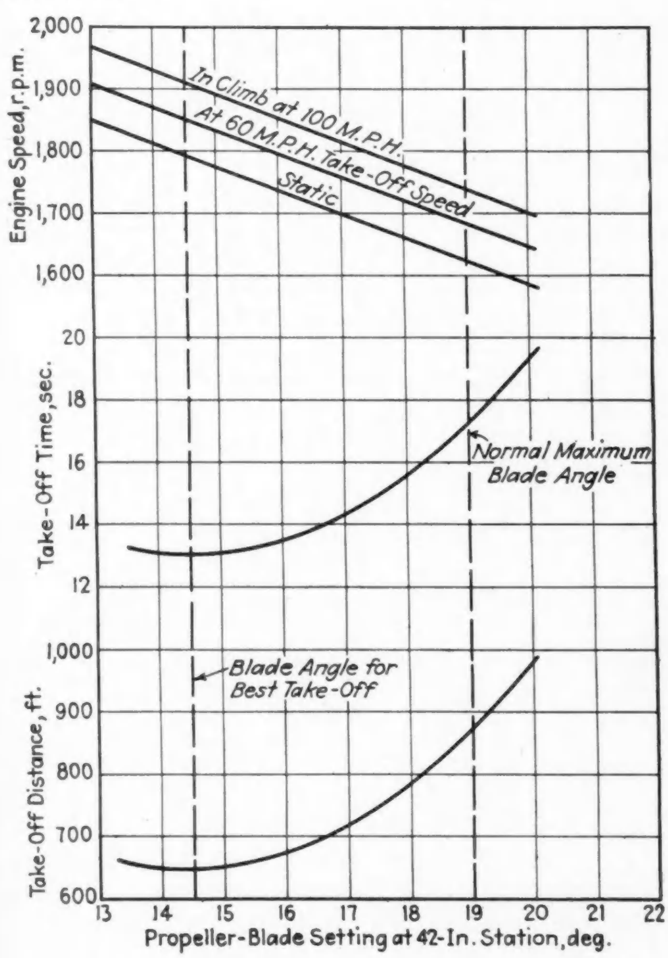


Fig. 15—Curves Showing Take-Off Characteristics at Different Propeller-Blade Angles

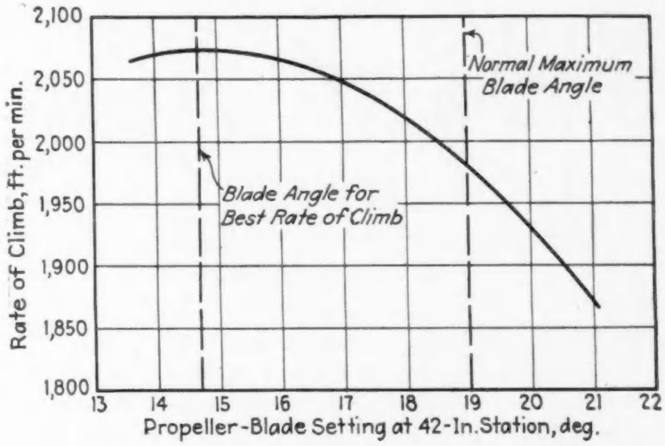


Fig. 16—Curve Showing Rate of Climb at Various Propeller-Blade Angles

Discussion

Kalman J. DeJuhasz[1]: — Determining the engine power in flight by measuring it directly with a suitable dynamometer undoubtedly would be superior to the inferential method based on manifold pressure and engine speed as mentioned in this paper. There does not appear to be any unsurmountable difficulty in constructing a suitable instrument for such purpose. A stiff resisting-member, which would allow a small amount of radial and/or axial displacement between the engine shaft and the propeller should be provided, and the small motion magnified by hydraulic, optical or electrical means. That a non-rigid coupling between engine shaft and propeller need not jeopardize safety and even may be beneficial, is proved by the Packard Diesel engine in which the propeller is driven through an elastic member. The necessary magnification of the elastic displacement could be well solved by means used in some high-speed-engine indicators. The problem would be greatly simplified in the case of geared engines in which the elastic measuring-element need not revolve but could be stationary.

During the war Professor Bendemann, in cooperation with the German Aeronautical Research Institute (Deutsche Versuchsanstalt fuer Luftfahrt), has developed an "hydraulic hub" which recorded, on gages placed in the cockpit, both the propeller torque and thrust. Later, the National Advisory Committee for Aeronautics also experimented with this instrument. According to my recollection, the tests proved the usability of the principle, though the arrangement of the hydraulic piping—which was carried from the front of the propeller, over the propeller and back to the cockpit in the shape of a large inverted U—was found somewhat cumbersome.

It would be illuminating if someone in closer connection with the aircraft end of this problem would state whether such an instrument is needed, what was wrong with the instruments tried out heretofore and what the requirements are for an instrument to be satisfactory. If there is a need for it, I am confident that the instrument end of the problem could be tackled successfully.

S. A. E. Transport Code Committee

THE accompanying code has been developed by the Automotive Transport Code Committee of the S.A.E. for the purpose of arriving at minimum-maximum limitations which can be supported on a strictly engineering basis.

It is presented as a matter of information and as a basis for discussion by Society members.

The Committee has therefore ignored considerations of policy or politics in framing this code, basing its recommendations entirely upon the facts of the case as established by published engineering research by qualified authorities.

The following notes are numbered, titled and lettered to correspond with the above code for convenience in reference.

1. Width

The maximum body width of 96 in. is recommended because, due to prevailing regulations throughout the country, this has come to be the accepted maximum, which has governed highway and bridge building practice with respect to width of roadway and side clearances.

The maximum width over dual pneumatic tires has been set at 102 in. in order to permit use of pneumatic tires, particularly those of the low-pressure type, of adequate size to support the axle loads contemplated in the code on vehicles now in service as well as on those which are now being and in future will be built for similar work. It was not contemplated to confine toleration of such widths to any stated period of time or to conversion of trucks already in service, because it has been found that the over-all width of dual tire equipment suitable for the maximum loads contemplated is such that, to confine over-all widths within 96 in., would necessitate such narrow spring centers as to endanger the stability of the vehicle.

It is furthermore to be realized that, with the restriction of body width to 96 in., the additional 6 in. of allowable width thus afforded the tire amounts to but 3 in. on a side, occurring below the line of vision of drivers of other cars so that it does not constitute an obstruction thereof. Furthermore, this additional 3 in. on a side does not impair safe clearance in connection with a 96-in. body inasmuch as the additional 3 in. is necessary as added clearance on the body to allow for sidesway; the tires being in contact with the road requiring no such additional clearance for sidesway. In other words, the necessary safe clearance between passing vehicles of 96-in. body width remains the same whether the tires are confined within the 96-in. width or overhang the extra 3 in.

Definition of the point at which measurement shall be taken is for the purpose of standardizing such measurement in correcting the inequalities in this respect as between states. The measurement is specified at the hub height both because this renders the measurements more easy to make and because it avoids measurement over the bulge at the point of road contact. This bulge, occurring close to the level of the road, does not constitute a clearance hazard and, inasmuch as some of the larger low pressure tires increase their cross-sectional width under maximum load at this point of bulge as much as 1¾ in., measurement at the bulge would serve to reduce the actual limit by this amount.

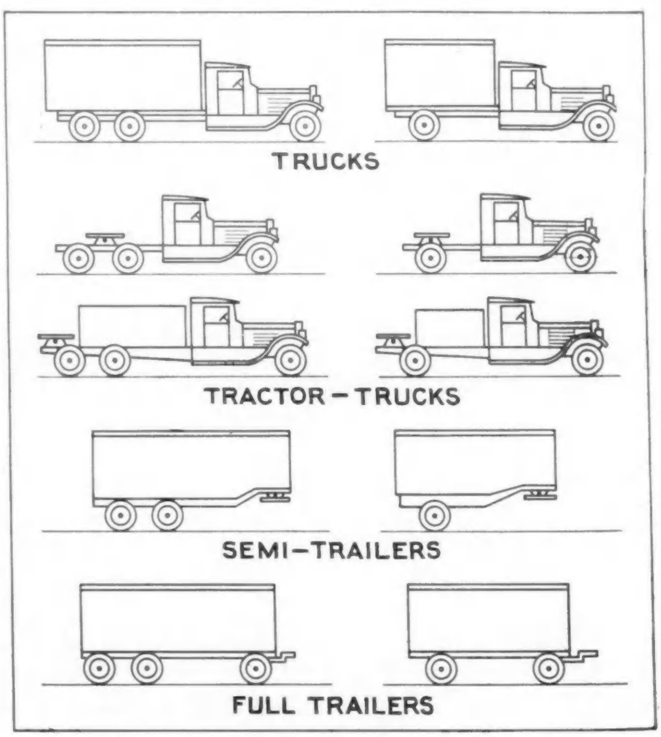
2. Height

This minimum-maximum height has been selected both because it is consonant with established precedent in the majority of states and because of the existence of a considerable amount of equipment which requires this amount of height and which could not readily be reduced in height. This measurement has become standard practice in the motor truck industry. The height is stated as unladen for the sake of definiteness, and because the unladen condition represents the maximum height.

3. Length

(*a*) *Classification of Vehicles.* These classifications have been set down as representing the accepted nomenclature of the industry and the standard nomenclature of the S.A.E. The only case in which any confusion might exist is in the case of tractor-semi-trailers. According to the recommendations of the Bureau of Public Roads submitted to the S.A.E. at its Transportation Meeting at Toronto, Canada, Oct. 5,

(*Continued on page 258*)



Makes Report for Study by Members

Load and Dimension Limitations on Motor Vehicles

Developed by S.A.E. Automotive Transport Code Committee

The following recommendations are based on practical engineering requirements for the design and operation of motor trucks and their combination of units.

1. Width

The maximum body width shall be 96 in. The maximum width over dual pneumatic tires measured on a line through the center of the hub, parallel to the ground, shall be 102 in.

2. Height

The maximum height shall be 12 ft. 6 in. when the vehicle is unladen.

3. Lengths

(a) *Classification of Vehicles*—Classification of single units for separate operation or for operation in combinations.

(1) Motor Truck. (A single self-propelled unit carrying its own load.)

(2) Tractor-truck. (A single self-propelled unit provided with a fifth-wheel for a semi-trailer and with or without a body for carrying its own load.)

(3) Semi-trailer. (A unit drawn by a tractor-truck by means of a fifth-wheel connection.)

(4) Trailer. (A unit drawn by a truck or tractor-truck and entirely sustaining its own load.)

(Note—Classifications to be accompanied by typical illustrations.)

(b) *Single Units*—The maximum length for any single unit shall be 35 ft.

(c) *Combinations of Units*—

(1) The maximum length of a combination of vehicles on all classes of thoroughfare 20 ft. wide or less, shall be 45 ft.

(2) The maximum length of all combination of vehicles on all classes of thoroughfares more than 20 ft. wide shall be 65 ft.

(d) *Special Equipment*—For single units over 35 ft. long and for multiple unit combinations of vehicles over 65 ft. long, special permits good for not over one year, shall be required.

(e) *Number of Units*—The minimum (or least) maximum number of units to be operated in any one combination of vehicles, shall be two.

4. Weights

(a) *Definitions of Thoroughfares*—

(1) Streets—Thoroughfares within municipalities and immediately contiguous metropolitan districts.

(2) Highways—Main arterial routes between cities and towns and connecting industrial areas.

(3) Roads—all others.

(b) *Weight Limitations*—The minimum (or least) maximum axle weight limitations in pounds, in lieu of limitations in gross weight and inch width of tires, shall be

	Streets (1)	Highways (2)	Roads (3)
High Pressure Pneumatics	22,500	18,000	16,000
Balloon Type Tires	22,500	20,000	18,000
Solid Tires (See note 1)	22,500	Not allowed	Not allowed

Note 1—Upon adoption of these weight regulations no new vehicles equipped with solid tires shall be registered and/or licensed for operation on Roads or Highways.

1932, and as adopted by the American Association of State Highway Officials, Nov. 17, 1932:

> "The truck-tractor and semi-trailer shall be construed to be one vehicle for the purpose of determining lengths."

To this the S.A.E. Committee was unable to agree, both because to consider such a combination as one vehicle for the purpose of determining length while considering it as two vehicles in determining the number of units in a combination is confusing and because this would restrict tractor-semi-trailers to 35 ft. in length. It appeared unreasonable to restrict the length of a tractor-semi-trailer combination to 35 ft. while permitting greater length in a truck and full trailer combination; particularly as this would operate to make the permissible body length on a tractor-semi-trailer less than on a straight truck, although its permissible load capacity would be greater.

(*b*) *Single Units.* The figure 35 ft. is so thoroughly established as the maximum length of a truck or bus, in existing state laws and regulations, in the various codes now extant and in the practice of the industry generally, that no deviation from this accepted standard has been contemplated.

(*c*) *Combinations of Units.* In this case the proposed limits have been divided into two groups, covering narrow and wide thoroughfares, respectively. This is in recognition of the fact that the length of a combination which is permissible on a wide thoroughfare is greater than that on a narrow one, inasmuch as passing is facilitated by the wider roadway, so that a longer combination of vehicles constitute less of an impediment to overtaking vehicles. It is instructive in passing to note that the Bureau of Public Roads recommendations are for 65 ft. flat and those of the American Association of State Highway Officials are for a flat 45 ft. The S.A.E. recommendations coincide with the latter for thoroughfares less than 20 ft. in width and with the former for thoroughfares wider than 20 ft.

(*d*) *Special Equipment.* Inasmuch as essential services in highway transport require the employment of certain special types of equipment and the hauling of indivisible loads which would exceed the above limitations, this provision has been included to correct an oversight which is generally made in other codes of similar nature and omissions sometimes made in state statutes which lead to confusion and difficulty.

(*e*) *Number of Units.* In considering the proposed limitations to two units, it should be borne in mind that this, as is the case with all other limitations proposed in this code, is a minimum-maximum, which is to say that it is intended to recommend that no states shall restrict the number of units in a combination to less than two. On the basis of actual experience, there seems to be no justification for confining motor transport to single units, whereas the advantages to be derived from the use of semi-trailers and full trailers are too well known to admit question. Doubtless in some regions where curves and hills are frequent and where roads are narrow, operation of combinations comprising three or more units is inadvisable. On the other hand, there are other districts comprising, as a matter of fact, the greater part of the Mississippi Valley and Great Lakes region, where three-unit combinations have been in safe and satisfactory operation for many years and where raising the limit to three units is entirely advisable.

It has been argued that, in view of the length limitations prescribed for combinations, it is not necessary to specify the number of units permissible. It has been felt, nevertheless, that, owing to the danger of snaking and skidding of a third unit in a train of three short units in mountainous country, the limitation on number of units should be retained. It is felt that to omit any reference to number of units would leave the way clear for forbidding more than one unit.

4. Weights

(*a*) *Definition of Thoroughfares.* As thoroughfares exist, fundamentally, as an adjunct to vehicular transport, it is believed that any classification should be according to the nature of the route involved from the standpoint of purpose rather than the kind or condition of the structure. As a general rule, the size of vehicle using various routes is governed by the amount and kind of tonnage to be handled, and this, in turn, is usually heaviest on city streets, next heaviest on main intercity highways, and lightest on secondary roads. Heretofore, the practice has been to classify weight allowances on the basis of arbitrary classifications as, Class A, Class B, and Class C roads. These classifications are almost invariably made by the highway departments with reference to the type of paving and general structure of the thoroughfare, without reference to its vital relationship to the transport requirements of the route. In the economic highway system road structures, widths, grades, and curvatures are determined by the requirements of the traffic normally flowing over them, and there is no more justification for uneconomically restricting the size of vehicles which may traverse a certain route, for the sake of preserving inadequate road structures which may form a bottle neck on a continuous route, than there is for laying a needlessly high type of pavement on a secondary route over which none but light vehicles habitually pass.

(*b*) *Weight Limitations.* In arriving at the minimum-maximum axle weight limitation contained in this code, the S.A.E. has necessarily been guided by the findings of the Bureau of Public Roads, which constitute the most scientific and best authenticated data on this subject extant. However, these recommendations of the Bureau's are confined to pneumatic tires and state roads. Thus it will be seen that the Bureau's recommendations are adhered to in the classification headed "Roads." As defined in the S.A.E. Code, "Highways" represent main arterial routes such as are commonly improved with higher type road structures than state roads generally. As the Bureau's own traffic investigation has shown, the use of heavier trucks in sufficient numbers to offer possibility of damage to lighter types of construction through fatigue effects occurs only on such arterial routes, and, in itself, constitutes an unquestionable economic justification for the construction of adequate roads. This is particularly true in view of the fact that the Bureau has shown that such strengthening of structure involves an increased cost of about 8 per cent.

Although none of the commonly accepted codes so far promulgated embrace city streets, it is well known that the main routes frequented by heavy trucks are almost invariably provided with pavements of great load-carrying capacity, particularly in view of the fact that speeds on city streets are necessarily slower than on country highways. It is also the general practice of cities to permit greater loads within their corporate limits than are allowed on state highways and a large number of the state laws exempt city streets from the provisions in their weight-limiting statutes.

Although the Bureau of Public Roads has made no recommendation with respect to axle weight limitations on solid tires, the S.A.E. Committee recommends a limit of 22,500 lb. on the basis of the experience of a great many years in our principal cities. Furthermore, in consonance with the accepted policy of the states, with the findings of the Bureau of Public Roads with respect to the comparative impact delivered by solid and pneumatic tires, and with the voluntary preference of truck operators, this Code contemplates the prohibiting of solid tires outside of cities and immediately contiguous metropolitan districts.

These limitations are expressed as axle weight limitations solely, because the investigations of the Bureau of Public Roads have proven conclusively that on highway structures it is wheel or axle weight and not the gross weight which determines potentialities for road damage. The Committee has chosen to express the limitations on a per-axle basis in preference to a per-wheel basis on account of the constantly shifting distribution of weight as between wheels on the same axle; because this definitely precludes any advantage being taken of axle constructions wherein more than two wheels run on a single axle; and because it prevents unfair or oppressive enforcement tactics such as would be possible under a per-wheel limitation where the weight test on individual wheels was taken on a crowned road.

Although mindful of the difference between the effects of gross weight as between highways and bridges, the Committee has not included the so-called bridge formula in its recommendations for weight limitations because it has been found that, in practical application, the bridge formula works out to so nearly the same results as obtained by the straight axle weight limitation method that no useful purpose would be served by complicating the regulation by the superimposition of this species of gross weight limitation thereon.

No recommendations are made as to weight-per-inch-of-tire-width limitations since, with pneumatic tires, no such limitation is necessary. The facts as developed by the tests of the Bureau of Public Roads are that any degree of overloading of a pneumatic tire which can be sustained by such tire without rupture tends to decrease rather than increase the impact delivered to the road.

The personnel of the Automotive Transport Code Committee was as follows: F. K. Glynn, chairman; B. B. Bachman, A. F. Coleman, A. H. Gossard, C. S. Lyon, A. F. Masury, E. S. Pardoe, C. A. Peirce, A. J. Scaife, A. W. Scarratt, Pierre Schon, F. L. Sage, J. F. Winchester.

Repair Problems of Marine Diesel Engines

As engineers we all know that every machine has to be repaired at some time or other, so we can best serve the interests of the Diesel engine if we look it over with a view to determining what sort of troubles it is subject to and how we can prevent those troubles or cure them if they occur.

The first thing we come to in the engine structure is the bed plate. There are troubles which arise from very remarkable flexibility. The flexing of the ship's hull will very often be reflected in the bending of the bed plate. I had a broken crankshaft which was due to a change in the hull alignment due to a certain disposition of cargo. The frame is subject to the same deflections of the bed. Very often deflections in frames will result through deflections in the bed. It would appear that for large engines it is not a very good aim to strive for maximum rigidity of frames. We want them to have a real flexibility. The small unit is a different proposition because bending moments that are made possible by pressures are comparatively small but when you get a very large engine, even a moderate-sized engine, the bending stresses are tremendous. If there is any deflection in the hull or any movement and if the frames are rigidly restrained, then something is going to give way.

Cylinder Liners

Then we come to cylinder liner problems. Troubles with liners are wearing and cracking. Wearing is a natural result of engine operation but there will be varying degrees of it and of course the kind of material has some influence on the rate of wear to the liners, but we can get very good liners nowadays and the liners we get today are far superior to what we got a few years back, so I do not believe that the material is of as much importance as the fuel. The fuel will vary as the amount of abrasive impurities contained in it, but I am convinced that all fuel should not be used in the engines without centrifuging.

Some years ago we used to talk in terms of five years as being the life of a liner. That doesn't mean a thing as you might have a ship tied up to the dock for five years if conditions continue as they are now. Liner wear in terms of years is a thing of the past. What we should do is to relate it to engine operation hours. In other words, how much does the liner wear per thousand hours of operation? So in attempting to establish some sort of standard on that basis I developed the fact a few years ago that 5/1000 of an inch per 1000 hours of operation was pretty good but it is a very interesting fact that today due to the improvement in liner material, that is very much too much. I have a number of records of ships in my collection today and from that it appears that 2/1000 of an inch per thousand hours of operation is something you can easily expect.

Cylinder Head Heat Troubles

The cylinder head is the most complicated casting on the engine and as such it suffers most from heat troubles. One of the not uncommon sort of repairs is the repair of cracked heads and that is always done by welding, either gas or electric. If the crack happens to be out near the periphery of the head then gas welding is satisfactory, but if it happens to be in a place where it is surrounded by unsymmetrical joints then we have to resort to electric welding. We "v" out the metal in the vicinity of the crack and then the sides of that "v" are drilled and tapped for steel studs and the studs arranged so they are in line. As the metal builds up the studs are completely imbedded in the weld. That is very satisfactory in service. If in doing that sort of thing we have to cross a valve seat with our weld it introduces some difficulty in the machining because when you are machining the valve seat the cutting tool will sometimes spring right over it, but the recently developed Tungsten carbide tools are proving very efficacious.

Breakage of crankshafts used to be considered one of the

fundamental features of Diesel engine operation, but those days are over. We have run across some broken crankshafts but they are the exception to the rule. The breakage of a crankshaft is always due to one of two things—either misalignment or torsional vibration. Crankshafts never break because of defective manufacture. If the alignment is right and there are no high order criticals in the operating range the shaft will run indefinitely. On a land engine you have a massive bed of concrete. When you once get it lined up it is likely to stap that way. On a ship the bed is always working and maintenance of alignment is not so easy. When it comes to torsional vibration we do not have the trouble we had in former days because that is a phenomenon that is pretty well understood and it is possible to calculate your criticals before the engine is built. Practically all engine builders have rotating masses analyzed before construction so as to take care of that point.

I ran into a very interesting thing in the Brooklyn Navy Yard. They have done some wonderful work in the matter of torsional vibration and I think they are just about as far advanced in their analyses to detect critical periods and ways of handling as anybody in the country. In their new engines they have been able to reduce their criticals to a single one around 220 revolutions and that is of a very low order. They have been able to neutralize one with another and by a clever disposition of rotating masses have been able to concentrate the straight torsional vibration in a breaker shaft. This is the idea used in rolling mills where tremendous stress is put on the driving shaft. There is a short breaker shaft in which criticals are centered and the torsional vibrations are centered so that if anything breaks it will be the short section of breaker shaft and the crankshaft will not be injured.

In connection with the subject of crankshafts, before leaving it I want to mention a feature that was very much neglected in earlier days—that was provision for removing this shaft if you had to take it out. I had to remove one shaft from a German engine and in order to get the bottom shells of the bearing out the whole shaft weighing 15 tons had to be lifted up 15 inches before we could take the bearings out. Of course, the right way to make those things is with round shells that can be rolled right out from under the shaft.

Now we come to the connecting rod. That is a part that suffers so infrequently that we can dismiss it without any discussion.

Crosshead troubles are normally limited to wear of heads. There have been cases where crossheads themselves broke.

The piston rod of a single acting engine does not give any trouble. Occasionally we have some trouble due to wear on the connecting rod attached to the crosshead. In double acting engines, on the other hand, piston rod trouble has been pretty frequent. In the early days of double acting engines breakage of piston rods was annoyingly frequent. Rods would break without apparent reason but several things have been found out about the design of the piston rods since that have done away with some of the trouble but the breakage of double acting piston rods has not stopped. The inside of the piston rods should be highly polished like a rifle barrel and then no corrosive substances should be allowed to come in contact with it. The theory is that cracks always start on the inside. If you have scratches, tool marks, etc., they will start cracks.

A factor in piston rod manufacture is the wear of the rods due to working in a stuffing box. It was thought that that was going to be a major problem and turned out to be no problem at all. I had occasion to measure up rods that had been chromium plated, on a vessel after twenty-eight thousand miles of operation and also the rods of another vessel that were not chromium plated. There was considerable difference. The ones that were not chromium plated wore considerably more than the others. That is a situation that does not exist today. Engine builders know better how to make packing that is suited for double acting engines so that wear today in double acting piston rods is no more cause for worry than the wear of a steam engine rod. Chromium plating is not necessary.

From the repairmen's point of view the piston is perhaps the most troublesome part on an engine. Pistons suffer from cracks. Cracking of pistons is due usually to a phenomenon similar to metal growth under heat and it is not uncommon. They are cracked through much as dried mud will crack in the heat. This is purely a heat effect. You may get cracks in the side of the piston walls. That is not a heat effect directly. Seizure of the piston is usually indicated when you have cracks in the piston walls but cracks in piston crowns can be repaired. It is a very common job, the procedure being similar for what I described for the head except that it is easier because welding can be used, but welding will not be successful unless you pre-heat the piston. On one ship I repaired all the pistons on the starboard engine and six months after they were in excellent condition. I do not know whether they are still running but I assume that they are. Associated with piston repair is the matter of metal growth. We get a surprising amount of metal growth.

Preventing Cooling System Repairs

Cooling system trouble usually is due to the lack of appreciation of what can be accomplished by preventive treatment. Many repairs on Diesel engines come in the class of unnecessary repairs—unnecessary in the sense that the reason for the repair should not exist and would not exist if proper care were taken of the engine. Cleaning of jackets, cleaning out of scale, sediment, oil, rust, should be just as much a part of routine of cleaning a Diesel engine as cleaning the boilers on a steam packet. It is best to establish a periodical routine for cleaning and also to take the same precautions against corrosion as would be taken with the boilers on a steamship. Fresh water can be seriously corrosive. We should make it a point to maintain cooling water in non-corrosive condition.

Engine repair costs are not comparable because we cannot get comparative figures. People who operate motor ships have some good figures but they will not give them to us. We can only say that Diesel engine repairs when they have to be made usually cost more than steam engine repairs but they do not have to be made so often if the engine is operated right.

The development that has been forced on the steam plant by the Diesel engine has brought about a very interesting situation.

In the efforts to reduce the fuel and steam consumption and to increase the economy, we have seen all sorts of devices introduced—superheaters, feed water heaters, etc.—so that modern turbine-driven steamships are complicated affairs. That is very good. It has brought down fuel consumption of a steamer but the plant is becoming more and more complicated, and on the other hand the Diesel engines are getting simpler and simpler all the time.

The Diesel engine of today is far simpler than the one we used to run years ago.

Excerpts from a paper read to Metropolitan Section, November, 1932, by Louis R. Ford, editor, Motorship.